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(Continued on page 35)

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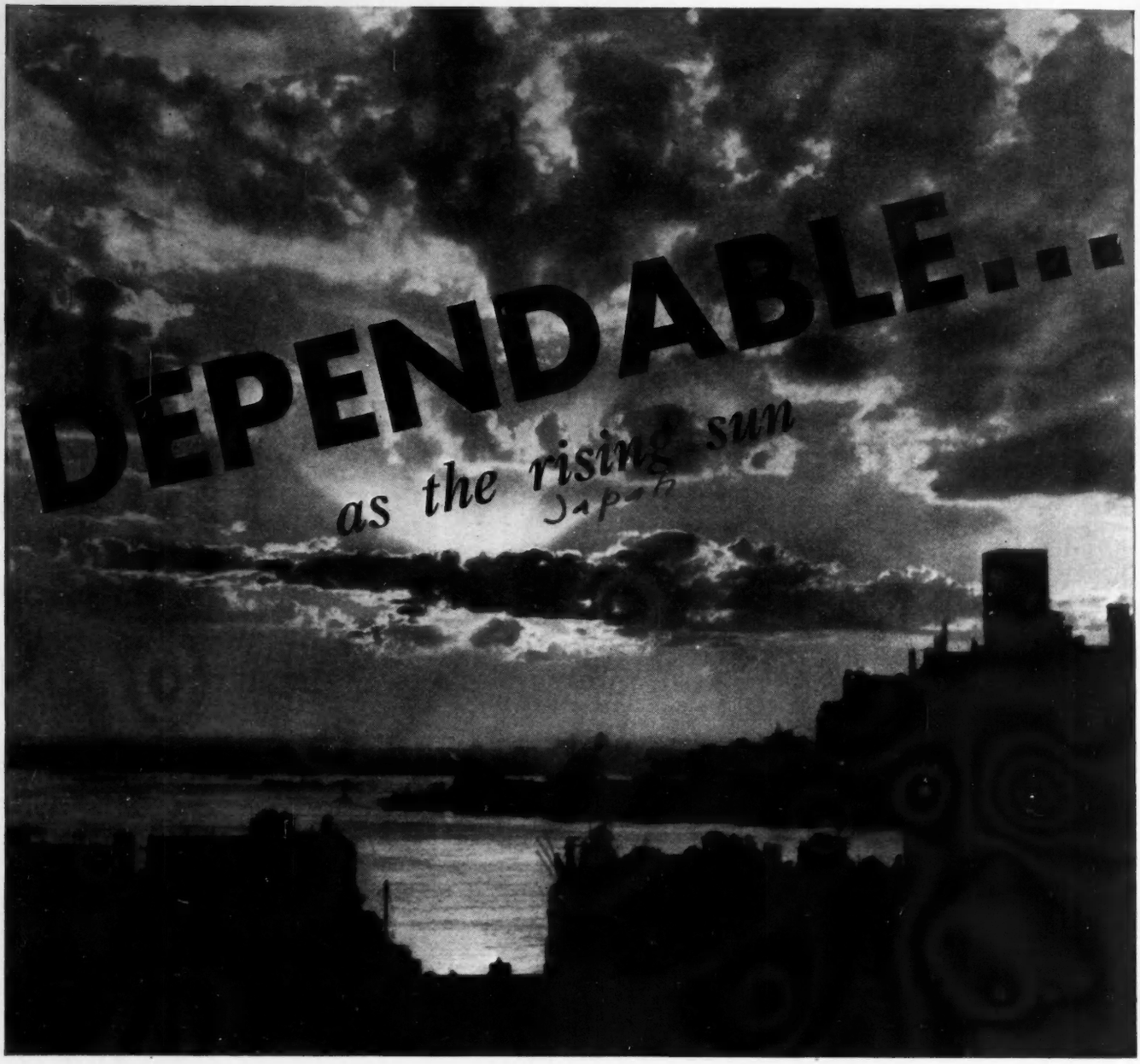

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UNITED STATES STEEL

Oil Filters in Public Utility Fleet Operation

By James I. Clower

Associate Professor of Machine Design, Virginia Polytechnic Institute

THIS paper has to do mainly with the care of the oil while in service, which care is considered as important, or more so, than its selection. The author points out that at present there is no reliable scientific method or test for determining whether or not an oil is suitable for further use, and that oils do not wear out but simply become contaminated with various impurities.

Impurities that tend to produce changes in the characteristics of crankcase oils while in service are considered under headings of inherent impurities and extraneous or foreign impurities. A number of types of filters are discussed with the various types of elements in use. Desirable features of a good filter are enumerated.

Although admittedly a controversial issue, the author believes that filters are decidedly advantageous and economically justifiable especially from the standpoints of fewer oil changes, decreased oil consumption, and reduced bearing and cylinder wear.

A discussion of the time when a filter element should be changed concludes the paper.

FLEET operators generally recognize that motor vehicles must be not only well cared for mechanically but also correctly lubricated in order to insure minimum cost per unit for safe and dependable operation, or in other words, that correct lubrication plays an important part in economical fleet operation. Let it be understood that correct lubrication means not only the selection of the most suitable oil for the vehicle at hand but also the use and care of the oil while in service. I am afraid some operators take it for granted that correct lubrication implies only the selection and that the problem is solved when the most suitable oil has been selected. This paper has to do mainly with the care of the oil while in service which, in my opinion, is as important or more so than selection.

Practically ever since the advent of the motor vehicle, it has been observed generally that crankcase oil undergoes during use both physical and chemical changes. This factor is of great importance because the useful life and operating cost are largely dependent on maintaining the oil in such a condition as to render effective and economical lubrication at all times.

The most generally accepted method of circumventing oil changes has always been to drain the crankcase after either a specified number of miles of operation or period of time. For obvious reasons, there has never been nor is there at the present time a common agreement among the interested parties as to the mileage beyond which it is not advisable to use oil. Not many years ago it was common practice to drain crankcase oil at intervals of 500 miles; today, the general practice is to drain at 1000-mile intervals. The fact is, however, that at present there is no reliable scientific method or test by which it is possible to determine whether or not an oil is suitable for further use. Considerable research has been conducted by various individuals, companies, institutions, and others in an endeavor to establish a reliable criterion for judging the useful life of crankcase oils, but the success thus far achieved has not been outstanding, although the effects of service on the properties of crankcase oils have been pretty well established at this time.

Oil technologists and other experts have known for many years that lubricating oils do not wear out, as many laymen believe. They simply become contaminated with various impurities, the removal of which renders the oil suitable for further service. Some authorities hold that properly reclaimed oils are better lubricants than are new oils, and the results of some tests support this claim.

Before attempting a discussion of the devices that have been produced for purifying oils while in service, it may be helpful if we consider somewhat in detail the factors or elements that tend to produce changes in the characteristics of crankcase oils while in service. These factors or elements are generally spoken of as impurities.

Sources and Nature of Impurities

For the purpose of discussion, impurities commonly found in crankcase oil may be grouped into two major divisions:

(a) *Inherent impurities,* which include substances (1) that are chemically formed in the oil while in service; (2) those that are present in the crude from which the finished oil is made and have not been removed completely during the refining process; (3) those that are introduced in the refining processes and are not eliminated completely in the finished product.

[This paper was presented at the Transportation and Maintenance Regional Meeting of the Society, Baltimore, Md., April 15, 1937.]

(b) *Extraneous or foreign impurities,* which get into the oil either during storage or while in service.

I believe that the majority of lubricating experts will agree that, if an oil, regardless of the base and refining methods employed, is placed in a perfectly clean crankcase of an engine and if no outside impurities are allowed to enter, it would still undergo some changes. Obviously, then, impurities are either present in the original oil or else develop while in service. Doubtless, both conditions exist. Crude oil contains many impurities, such as water; various kinds of solid particles; complex sulphur compounds; oxygen, nitrogen and other gases; and some unstable, complex hydrocarbons which oxidize readily and hence must be considered impurities.

In the process of refining the oil frequently is mixed intimately with various solvents, acids, alkalies, water, and air.

Regardless of the care exercised in the refining processes, some of these so-called inherent impurities remain in the finished oil. The extent to which they are present in the finished product, of course, depends on the care exercised by the manufacturer. Today no reputable refiner permits his product to leave the plant containing free acid, alkalies, water, and so on, and it is safe to say that, when oil is purchased from a reputable manufacturer, it is practically free from such impurities. By far the most serious and important inherent impurities are those that develop through chemical changes while the oil is in service. In general these are products of oxidation (commonly called asphaltenes) and are derived from the unstable hydrocarbons. The fact must not be lost sight of, however, that, although most oils are practically free from refining chemicals, yet a mere trace of any one of these chemicals may, in the presence of extraneous impurities, act to cause excessive oxidation and sludging.

There are three chief sources of foreign impurities: (1) from the air taken through the carburetor and the crankcase ventilating system; (2) from the combustion of the fuel charge and generally a small portion of the lubricating oil; and (3) from the interior of the engine itself.

Atmospheric air is never clean. Impurities which contribute to atmospheric pollution include carbon from the combustion of fuel, particles of earth, sand, ash, animal excretion, stone, wood, rust, paper, threads of cotton, wool, and silk, bits of animal and vegetable matter, and pollen.

A microscopic examination of the impurities in city air shows that a large percentage of the particles are carbon. Atmospheric impurities are generally classified according to the size of the particles, as dusts, fumes, and smoke. Dusts are particles of solid matter varying from 1.0 to 150 microns in size. Fumes include particles resulting from chemical processing, combustion, explosion, and distillation, ranging from 0.10 to 1.0 micron in size. Smoke is composed of fine soot or carbon particles, less than 0.10 micron in size, which result from incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco. In addition to carbon and soot, smoke contains unconsumed hydrocarbon gases, sulphur dioxide, sulphuric acid, carbon monoxide, and other industrial gases capable of reacting with the unsaturated hydrocarbons of oils.

Obviously the character of air impurities varies widely with the locality; however, the following are approximate averages of atmospheric dust concentrations:

Rural and suburban districts	0.2 to 0.4 gr. per 1000 cu. ft.
Metropolitan districts	0.4 to 0.8 gr. per 1000 cu. ft.
Industrial districts	0.8 to 1.5 gr. per 1000 cu. ft.

On the average, approximately 1000 cu. ft. of air are required for combustion purposes alone per gal. of gasoline. To this volume must be added the quantity of air used for crankcase-ventilating purposes.

Thus it will be appreciated that, in the course of the life of a motor vehicle, a considerable quantity of atmospheric impurities enters an engine, notwithstanding the fact that all present-day engines are universally equipped with efficient air cleaners; at least, they are efficient when new and when properly cleaned in service.

Combustion impurities consist of carbon particles, sulphur and its compounds, gum, water, carbon dioxide, and unburned fuel. Most of these impurities, of course, are ejected from the engine through the exhaust but some, regardless of the mechanical condition of the engine, escape or work their way past the pistons and rings into the crankcase oil, thereby contaminating it. The extent to which this class of impurities contaminates the oil depends mainly on the mechanical condition of the engine, especially on the effectiveness of the rings to prevent blowby, the conditions of the ignition system, characteristics of the fuel, and other factors.

It is common knowledge that a considerable portion of the unburnt fuel works its way past the rings and enters the crankcase, creating what is commonly termed dilution. Not many years ago, dilution was given frequently as one of the major reasons for crankcase draining. Since the introduction of the crankcase ventilator and thermostatic control of the cooling water, we have been hearing less and less about crankcase dilution. It is questionable whether or not dilution has merited the extensive attention that it has received in the past. It is now generally believed that it is not, except in a few cases where operating conditions are unusual, necessary to drain the crankcase because of excessive dilution. It has been said that dilution permits rapid wear. There do not appear to be sufficient test data to substantiate this conclusion but, on the contrary, recent test results indicate that no appreciable increase in wear is discernible even with extremely highly diluted oils. Since it is generally believed now that too heavy oils have been used in the past, in all probability dilution has, in the past, done more good than harm by reducing the viscosity of the oil, thereby insuring quicker distribution to the various surfaces requiring lubrication.

In the southern section of the country it is questionable whether it is ever necessary to change oil because of excessive dilution. In the northern sections of the country, if a vehicle is used during extremely cold weather for only short trips when frequent starts and stops are made and the choke is used freely and, especially if the motor is in poor mechanical condition, one may be justified in changing oil because of excessive dilution. It should be remembered, however, that our Society recommends 10-W oil plus 15 per cent kerosene for temperatures lower than —15 deg. fahr.

It is a known fact that diluent is present in a varying degree in all crankcase oils, and that the percentage present varies with the type of engine, its mechanical condition, its manner of operation, and especially with the atmospheric condition. Dilution, by its very nature and volatility, reaches an equilibrium wherein the rate of its evaporation and that of its formation are equal. Equilibrium is generally established after only a few hundred miles of operation and frequently in less than one hundred.

In the more or less heavy-duty service of motor vehicles the percentage of dilution is relatively small, generally ranging from 1 to 7 per cent.

Impurities that originate within the engine itself consist mainly of small metal particles worn from the engine, core sands, drillings, metal chips loosened by the hot oil, vibration and metal expansion and, in some cases, iron oxides resulting from the corrosion of various interior surfaces.

As prevention is always preferable to cure, it should be the first concern of the operator to prevent as many of these various impurities from entering the crankcase as is possible.

First, he should only use oil of a high degree of purity. Second, he should make certain that both the carburetor and ventilating air filters are of an efficient design and that they are properly cleaned and serviced periodically. Third, that the engine is maintained in good mechanical condition at all times. The practice of preventive maintenance will do much to make the last measure effective.

Perhaps none of these impurities exists in crankcase oil individually in a sufficient quantity to cause serious trouble. So it is not the individual impurity acting alone but rather the combined action of all of them that may cause serious trouble. Moreover, it is more likely that it is their mutual effect on one another that causes trouble. Undoubtedly some of them combine chemically to form entire new compounds while others mix mechanically. It is this combined chemical and mechanical mixture that is generally called sludge, the character of which varies widely, depending on both the original oil and the extent to which it has been exposed to foreign impurities. It has been proved that impurities promote and accelerate oil oxidation and sludge formations.

In spite of all precautions that an operator might take, impurities will gradually accumulate and eventually reach a point where the oil becomes unfit for further use unless means are provided for their removal. This condition, as previously pointed out, is not due to the oil wearing out, but because it has become contaminated with these detrimental impurities, which are likely to clog oil ducts, grooves, passages, and pipes as well as to cause ring- and valve-sticking and excessive wear. Sludge is usually of an acid nature and, if permitted to collect for any appreciable period, may cause corrosion especially to the highly polished surfaces of the barrel, piston, rings, rods, and crankshaft.

It is evident that dirty oil cannot and does not lubricate as satisfactorily as does clean oil. It has been appreciated ever since the very early days of motor vehicles that the removal of these impurities is essential in order to avoid lubricating troubles.

Figs. 1 and 2 are photomicrographs (X443) of oil removed from the crankcase of a passenger vehicle after 1000 and 3763 miles respectively of operation. They clearly show the effect of service on the quantity of impurities present. It will be noted that the particles in Fig. 1 are very small and separated whereas, in Fig. 2, they have formed agglomerates.

Figs. 3, 4, 5, and 6 show the effect of service on some of the properties of an oil used in a passenger vehicle. From Fig. 3 it will be noted that the film breakdown pressure increased with use and, in all cases, was greater for the used than for the new oil, reaching a maximum of 7000 lb. per sq. in. at the end of 350 miles of operation. The hemacytometer count shows practically a uniform increase, reaching a maximum of 13,605,000 particles per cu. mm. after 3763 miles of service.

Shown in Fig. 4 are the neutralization and precipitation numbers of the same oil. Both of these curves show a more or less gradual increase as should be expected. Likewise, as shown in Fig. 5, the ash and iron content and carbon residue show increases.

As to be expected, and as shown in Fig. 6, there is no definite trend of viscosity and dilution. There is, however, a rather definite correlation between viscosity and dilution. In practically all instances where viscosity increases, dilution decreases and vice versa. This relation is, of course, as we should expect it to be and demonstrates that changes in viscosity are caused mainly by dilution.

From the foregoing illustrations and discussion it is obvious that any refiner would, to say the least, have considerable difficulty in recognizing his own product after it had been subjected to a few hundred miles of service in the average crankcase of a motor vehicle. Moreover, it hardly appears necessary to say that some device constructed so that it will tend to remove the various impurities would be a decided advantage in motor-vehicle operation.

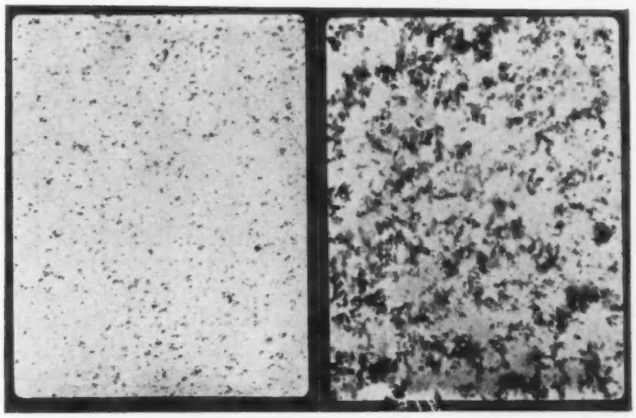

Fig. 1 – Photomicrograph of Crankcase Oil Drained from a Passenger Vehicle after 1000 Miles of Operation (X443)

Fig. 2 – Photomicrograph of Crankcase Oil Drained from the Same Passenger Vehicle as in Fig. 1 after 3763 Miles of Operation (X443)

Types of Oil Filters

At the present time there are available several different oil-filtering devices especially designed for motor vehicles. Time does not permit a detailed description of these devices.

The heart of any filter is obviously the element itself, and it is safe to say that practically all materials known to possess filtering characteristics have been used or are being used at present. Some of these materials are cotton waste, slag wool (acid), slag wool (alkaline), wool felt, Canton flannel, chemically treated paper, Fuller's earth, cellulose fiber, rock wool, brewer's filtermass, felt discs and, doubtless, others. In some filters two or more of these materials are used in combination.

It will suffice to say that, in the case of the early filters, the idea was to filter out the abrasive, solid impurities and sludge after it had formed. In general, they were simply rather efficient strainers of the felt-disc and screen types; frequently they were too small and consequently became clogged after a comparatively few miles of operation unless they were cleaned frequently. Moreover, in general, the job of changing the element was tedious and time-consuming. I do not believe that anyone will deny that most of the early filters had many shortcomings and, in consequence, were generally discredited. In all fairness to these early filters it should be mentioned that one of the major indictments against them was that they did not remove the dilution. As previously explained, this problem has been solved largely by other means, such as crankcase ventilators and thermostats.

Many of the early filters were not capable of removing the very fine carbonaceous material which is mainly responsible for oil discoloration. This inability to maintain color-free oil over a reasonable service period likewise caused many operators to condemn oil filters.

Whereas filters were originally designed to filter out sludge after it had formed, today the idea is to remove the sludge-forming elements before they have had an opportunity to combine. This change of viewpoint in regard to filter design has done much toward the improvement of filters during the past five years.

Although in the past many types of filters have been experimented with, the present trend appears to be toward the use of the bypass, removable-element type which combines both the principles of filtration and absorption.

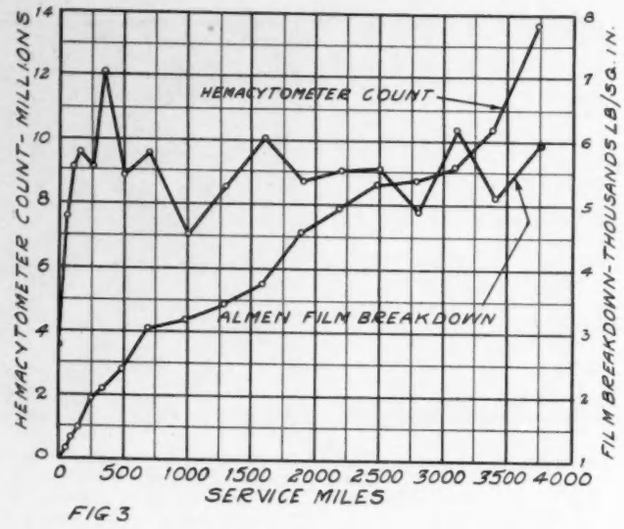

FIG 3

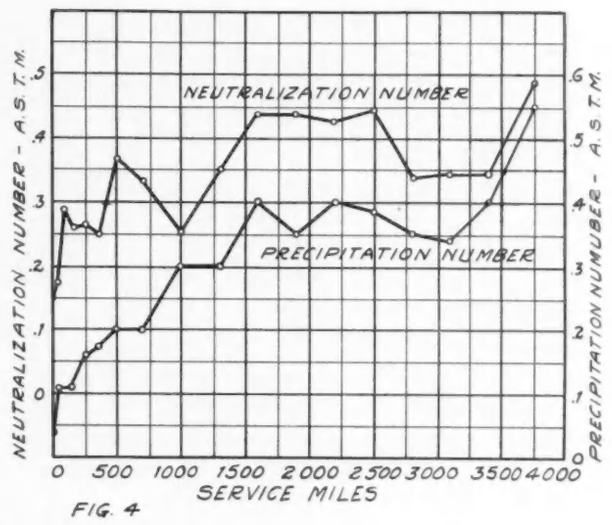

FIG. 4

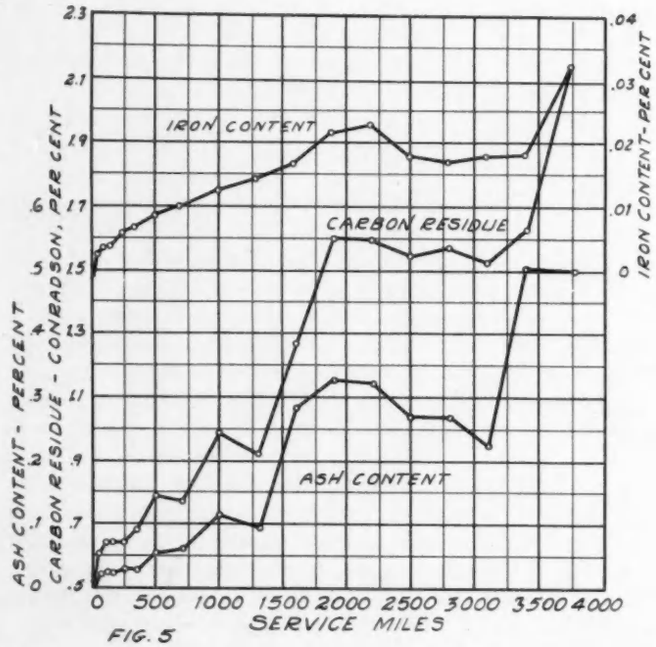

FIG. 5

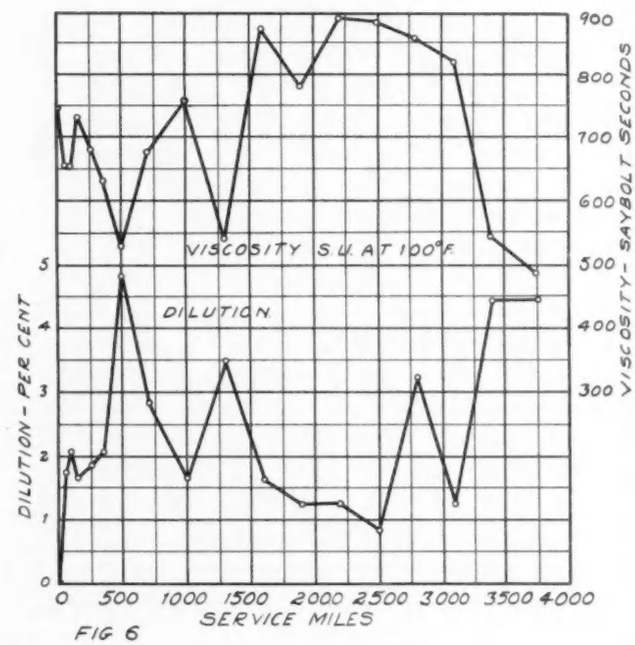

FIG 6

Figs. 3-6 – Effect of Service on Various Properties of an Oil Used in a Passenger Vehicle

Figs. 7, 8, 9, 10, and 11 show some of the modern designs of this type of filter.

Shown in Fig. 12 is one design of a metal-element type which is self-cleaning. In this design corrosion-resisting wire is wound in threads cut on the outer surface of a heavy brass cage. The minute spaces between the coils of wire constitute the mesh through which the oil is filtered. By varying the diameter of wire, varying degrees of filter fineness are provided. Spacing as small as 0.0004 in. may be secured in this manner. Other metal-element filters employ metal discs and ribbon.

Fig. 13 shows the accumulation of sludge on the outer surface of the element of the filter shown in Fig. 11.

Most operators prefer the bypass type in which a small percentage of the full discharge from the oil pump passes through the filter each oil cycle, in preference to the full-flow type in which the full discharge from the oil pump passes through the filter.

I am not in a position to give information regarding the efficiency of any particular make of oil filter, but believe the following discussion of the desirable features of a good filter may be of assistance in the choice of a suitable oil filter for any particular purpose. To expect to find any one filter possessing all of these desirable features, would, I am afraid, be overly optimistic.

Desirable Features of a Good Filter

From the foregoing discussion it is obvious that a good filter should possess a maximum number of the following desirable features:

(1) It should remove all harmful impurities, including asphaltenes and the so-called "colloidal" carbon.

(2) It should be convenient to service without disturbing the oiling system.

(3) It should be adapted easily to all makes of engines.

(4) Its first cost and maintenance cost should be such that its use is economically justifiable.

(5) It should be reliable and present no oil-loss hazard.

(6) It should be capable of maintaining its efficiency for a comparatively large number of miles, say 5000, or over a considerable period of time.

(7) It should maintain an acceptable oil color for a reasonable period of operation.

(8) It should be sufficiently sturdy to withstand the rough service to which it is subjected in motor-vehicle operation.

(9) The filtering element should possess characteristics such that it will not pack or channel.

(10) The volume of oil passing through the filter should be controlled so as not to wash back into the oil stream impurities collected.

(11) The construction should be such that the dirty oil cannot "short-circuit."

(12) The "sump" should be of ample size to collect the heavier solid impurities and water.

No attempt has been made to arrange these desirable features in the order of their importance.

The importance and desirability of these features are, I believe, self-evident.

Possible Benefits

Varied opinions are held as to the benefits likely to accrue from the use of an oil filter on a motor vehicle. No two people appear to agree on this question; I suppose, because it is very difficult if not impossible to obtain comparative data. So many other influencing factors are present that it is very difficult to eliminate, in the case of a test, all of them excepting that of the filter itself.

However, although actual cost figures are not available, it appears to many that filters are decidedly advantageous and economically justifiable. Many operators are prone to measure their advantages in terms of their effect on oil changing only. In this connection, it appears certain that the use of oil filters greatly extends the useful life of oil. I have not yet convinced myself that drain plugs should be eliminated from crankcases, although some filter manufacturers advertise that their filters eliminate the necessity for oil changing. Assuming that, without the use of filters, drains are necessary at approximately 1000-mile intervals, it appears that this interval can be safely tripled or quadrupled when efficient filters are used. Many large fleet operators are operating 3000, 4000, 5000, 6000, and more miles between drains. Some operators never drain except for the purpose of changing from summer to winter grade and vice versa.

The effect of the use of filters on oil consumption is a much debated question. Perhaps not directly, but certainly indirectly, filters have a favorable effect on oil consumption. Contaminated crankcase oil undoubtedly promotes sticking of both compression- and oil-control rings and causes restriction of oil flow and, in severe cases, complete stoppage of flow through drain slots in the rings and drilled holes in ring grooves. Manifestly, any interference with the free flow of oil through these vital passages tends to increase oil consumption and, conversely, anything that tends to lessen the cause of their becoming plugged is likely to reduce oil consumption.

Decreased wear is another benefit that is difficult to evaluate and one on which sharp differences of opinion exist. Bearing wear and especially ring-and-bore wear are influenced by so many factors that it is practically impossible to isolate the effects of any one factor. Some of the known factors influencing wear are jacket-water temperatures, hardness and finish of cylinder walls and rings, the design of the piston assembly, character of operation, character of fuel used, air-fuel ratio, viscosity of the crankcase oil, quantity and character of impurities that enter the induction system, design and capacity of the lubricating system, and perhaps others. It is my opinion that some of these factors, or perhaps a combination of two or more of them, have a greater wear effect than the maintenance of the oil in a practically impurity-free condition. In other words, excessive wear may occur under abnormal conditions irrespective of the oil-filter efficiency and the resulting cleanliness of the oil. On the other hand, if all other conditions are normal and equal, then clean oil must naturally minimize wear. Some filter manufacturers claim that their filters are very effective in minimizing wear at all points in the engine. Dependable data on this point appear to be lacking, as it is very difficult to secure operating-cost records that furnish comparisons over a representative number of units. Many cases are cited of individual units, however, which strongly indicate reduced wear through the use of filters.

On this point a chief engineer writes: "We feel certain that the application of this type of filter has been responsible for increasing the life of our cylinder bores and other engine parts but, due to a change in material and hardness of our blocks during the time when we changed our production filter specifications, we have no records of the actual improvement due to the filter alone.

"We recently had occasion to check a group of engines, all of which had been in service 80,000 miles. The bores were measured and averaged 0.0035 in. oversize at a point 1⅛ in. down from the top of the block, and at a point 5 in. down from the top of the block, the diameter measured 0.0015 in. oversize. New rings were applied and another 50,000 miles of operation were obtained, and still reboring was unnecessary. The rings were perfectly good and free at the time of the 80,000-mile inspection, but we felt it would be practical to replace them while the engine was down."

A superintendent of a comparatively large fleet offers these comments in regard to wear: "In a fleet operation it is difficult, in fact impossible, to relate wear to any single item of equipment or operating practice. We know that sludging and gumming, since our adoption of filters and proper scheduling, is relatively infrequent. How much of that is due to the oil (refined), filters, non-change policy, warm-engine operation (thermostats, covers, and so on) mechanical condition, and so on, no one can possibly determine."

An executive of one of the larger coach and truck manufacturers in this country comments as follows: "In any engine operating under conditions which indicate a filter to be desirable, it is safe to say that slower cylinder and ring wear and less ring- and valve-sticking will be experienced with, than without, an efficient and properly maintained filter."

Still another member of a large truck and coach firm says: "By keeping the oil free of colloidal graphite, metal particles, and other foreign matter, it retards bearing and cylinder-wall wear."

Some filter manufacturers mention improved fuel economy as a benefit derived from the use of their filter. In my opinion, the only effect the use of a filter could possibly have in regard to gasoline consumption would be through maintaining the engine in a somewhat better mechanical condition and thereby effecting higher thermal and mechanical efficiencies. This possible saving of fuel, I do not believe should be considered of major importance nor sufficient reason to install oil filters.

The cost items of repairs and depreciation are closely associated with wear, both being proportional to the wear. It is obvious, since these two items constitute an appreciable portion of the total operating cost, that any, however small, reduction in either of them is of appreciable value in dollars and cents. Personally, I am of the opinion that the use of filters is justified from this standpoint alone.

Occasionally, one hears the complaint made that filters do not reduce dilution of the crankcase oil. This indictment, I do not believe to be a serious one. As has already been said dilution as a problem is, in my estimation, a relic of the past, perhaps with the exception of a few cases where operating conditions are unusual. Although admittedly most filters will not reduce dilution directly; yet indirectly they do tend to establish a dilution equilibrium percentage lower than that which would prevail were no filter used because of the tendency of the filter to maintain the engine, especially the piston

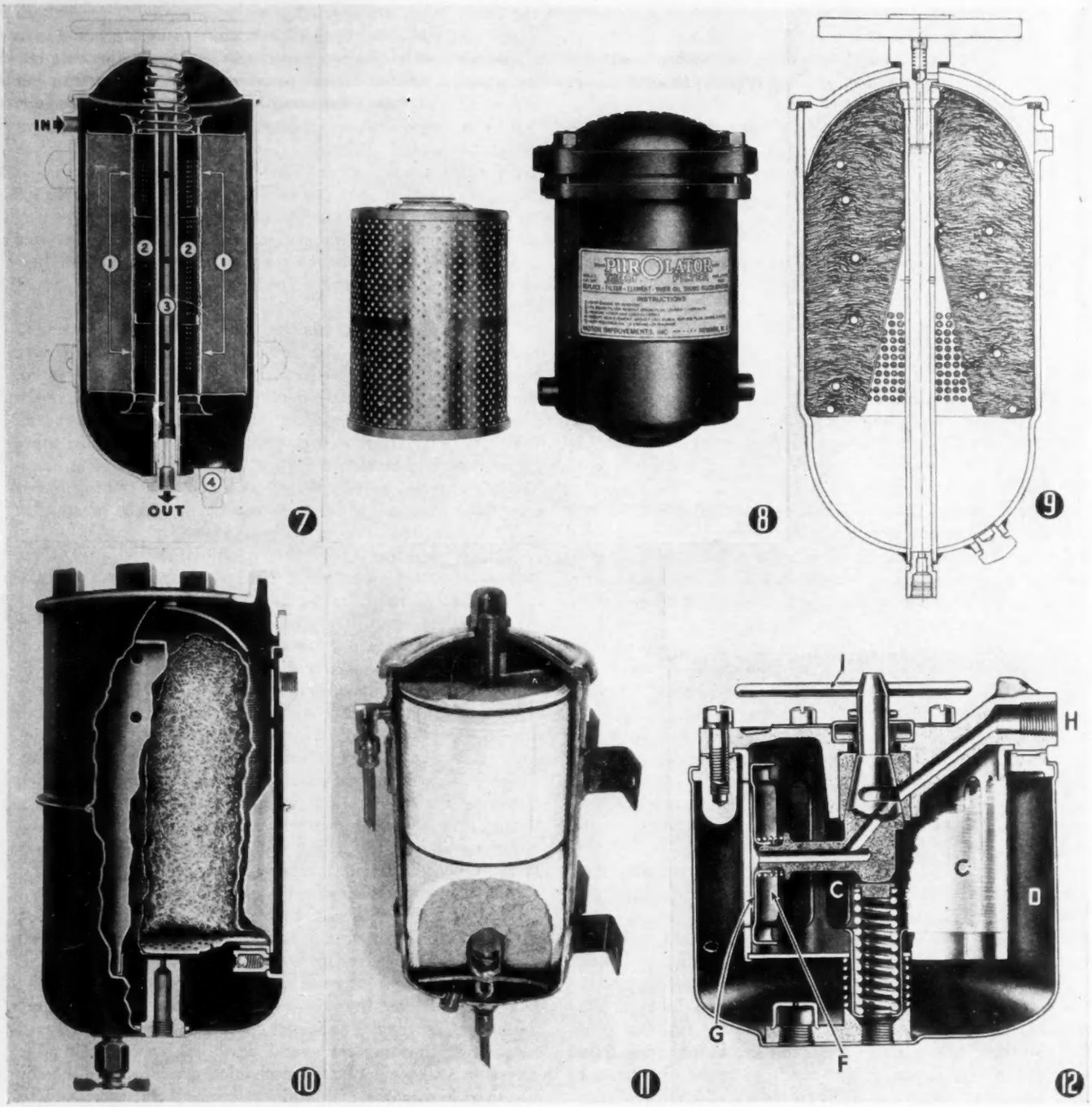

Fig. 7 – Renewable-Element, Bypass Filter Which Employs the Principles of Filtration and Absorption – The Element Is a Rock-Like Material Called Igneonite

Fig. 8 – Renewable-Element, Bypass Filter Having a Vegetable-Product Element Supported by a Perforated-Metal Housing

Fig. 9 – Renewable-Element, Bypass Filter Having a Specially Treated Cotton-Waste Element Supported by a Spring Which Prevents Packing

Fig. 10 – Renewable-Element, Bypass Filter Which Employs Two Different Filtering Media

Fig. 11 – Renewable-Element, Bypass Filter Having a Molded Fuller's Earth Element Wrapped with Five Layers of Cellulose

Fig. 12 – One Design of a Metal-Element Type Filter Which Is Provided with a Self-Cleaning Mechanism

and rings, in good mechanical condition over a longer period.

Still another possible benefit that points toward the desirable use of a filter is the matter of oil temperature. It has been observed that the oil temperature is generally decreased by the installation of a filter. Advantage may be taken of this decreased temperature to use a less viscous oil and hence realize less starting difficulties. Also the use of less viscous oils would be beneficial in that they, in general, other things being equal, tend to sludge less.

Also should be mentioned the possible savings resulting from the decreased time out of service for repairs. It should be remembered that the total cost of repairs is not simply the direct labor and parts costs, but also includes the loss in revenue due to idleness, which may be an appreciable amount in some cases.

In brief, the possible benefits accruing from the use of filters may be summed up by saying that operators may expect longer vehicle life, lower repair cost, and lower total cost of operation. It is a matter of worthy and careful study by all operators and is sure to return worthwhile dividends. Although the day is not here now, I believe that it will come when we will not change oil except to meet seasonal demands.

When Should Element Be Changed?

No fixed period of time nor definite number of miles of operation can be set that will be satisfactory for all cases. The renewing of the filtering cartridge depends on many factors, such as (1) the characteristics of the oil being used; (2) condition of the engine; (3) operating conditions; (4) climatic conditions; and (5) the character and capacity of the filtering element itself. Obviously, these factors vary widely from fleet to fleet and, furthermore, within a single fleet, especially in the case of a mixed fleet. In general, it can be said, I believe, with positiveness, that the filtering element should be capable of rendering effective filtration under normal conditions for at least 2000 miles. Many under rather severe conditions perform satisfactorily for from four to five times this distance. The average, perhaps, is 3000 to 4000 miles.

The most prevalent method used today by operators to determine oil life is by color. This method, I believe, is also universally recommended by the oil-filter manufacturers. It has the decided merit of being simple and easily performed by practically anyone; yet I doubt if many believe it is free of all shortcomings.

It is a recognized fact that the color of oil can be changed appreciably simply by heating. Everyone who has performed a flash-point test is aware of this fact. Not many, I am sure, will contend that the quality has been appreciably affected by the application of such a moderate amount of heat.

Committee D-2 of the American Society for Testing Materials has this to say with reference to color as an index of lubricating value: "Color requirements of lubricating oils are frequently overemphasized as color does not necessarily indicate quality."

Most of the discoloration which oil undergoes in crankcase service is not, of course, due to heat, but to the accumulation of the various impurities. In general, however, it will be found that, even after all impurities are removed, the oil is still discolored some. This discoloration is in no way harmful and should not cause concern. Personally, I am of the opinion that color-clear oil indicates commercially acceptable filtration. Certainly color-clear oil is abrasive- and moisture-free and offers assurance against clogged oil ducts, passages, and holes and protection against severe sludge formations.

It will be appreciated that, in the ultimate analysis, when to change the filtering element is reduced to answering the question when is an oil unfit for further service.

As pointed out in the introduction, there has not yet been developed a scientific test or combination of tests by which it is possible to determine whether or not an oil has rendered its maximum safe life. It is granted that an oil expert can, after a series of time-consuming and involved laboratory tests, render a fairly accurate decision. However, his decision is largely a matter of opinion and is usually based as much on experience as on scientific facts.

Even if present-day laboratory tests did definitely establish a safe criterion for determining oil-life, yet they are not practical from the standpoint of fleet operators because they require a considerable investment in test apparatus and the service of an experienced oil chemist.

From the standpoint of a fleet operator, any test for determining the useful life of an oil must be simple and capable of being performed by the average mechanic. Doubtless most fleet operators employ some method of determining the useful life of an oil. I doubt seriously if many of them are satisfied or convinced that their method is infallible. It seems to me that this is one of the most important problems in the operation of motor vehicles yet to be solved. Private owners are as much in need of the answers to this question as are fleet owners.

No one feels justified in assuming the risk of a costly repair job for the comparatively insignificant cost of an oil or filter-element change; yet there undoubtedly has been, and still is being, run to the sewer annually millions of miles of "oil-life." This waste although individually small is yet enormous in the aggregate. A conservative estimate indicates that more than 250,000,000 gal. of oil are dumped annually from the crankcases of our 28,000,000 motor vehicles. If crankcase drainings were to be extended from an average of 1000 to 2000 miles, more than 3,000,000 bbl. of refined oil would be saved yearly.

This condition is not only important because of the direct savings that would be realized by motor-vehicle owners, but also from the standpoint of the conservation of one of our most essential natural resources. The present estimated reserve supply of crude petroleum for the United States is approximately 12½ billion barrels. The recent yearly consumption has averaged 1 billion barrels. Consequently, if no new reserves were discovered in the meantime, our known reserve supply would become exhausted by 1950. During the past two years, new discoveries have just about kept pace with consumption. This year, the indications are that consumption will exceed new discoveries, notwithstanding the almost superhuman efforts that are being made by oil companies to discover new supplies. Certainly a major National undertaking, such as war, would seriously reduce our present reserve supply.

(*Continued on page* 392)

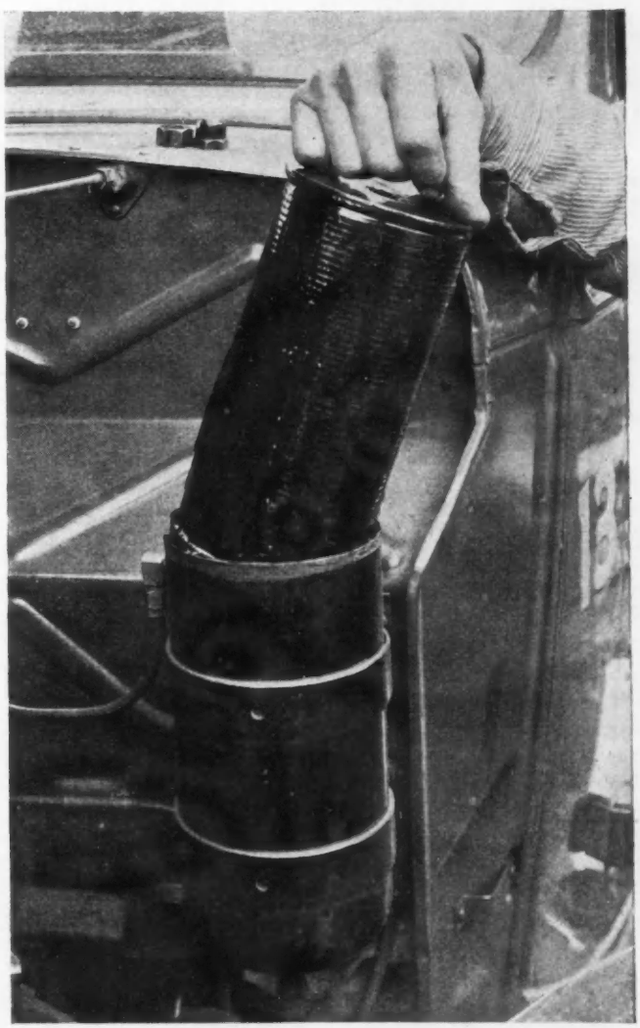

Fig. 13 – Installation of a Filter of the Type Which Is Shown in Fig. 11 – The Exposed Element Shows the Accumulation of Sludge on the Outer Surface

The Design of Metal Fins for Air-Cooled Engines

By Arnold E. Biermann

Junior Mechanical Engineer, National Advisory Committee for Aeronautics

THIS paper gives the results of an analysis made to determine the proportions of aluminum and steel fins to dissipate maximum quantities of heat for several pressure differences across a finned cylinder. The power required to force the cooling air between the fins and the relative weights of the various designs are presented.

The calculation of the heat flow in the fins is based on an experimentally verified, theoretical equation and the surface heat-transfer coefficients and pressure differences were taken from previously reported experiments.

In particular, the analysis concerns fin proportions for minimum pressure drop, minimum power, and minimum weight.

HEAT-LOSS measurements made on conventional air-cooled engines have shown that, for satisfactory operation, the heat dissipation from the cylinder walls is approximately proportional to the 0.65 power of the indicated horsepower developed.[1] Thus a relatively large increase in power is permitted by a small increase in the overall heat-transfer coefficient U. This result indicates the importance of improvements in engine cooling as related to the general problem of increasing the specific power output of engines.

The solution of fin-design problems is not only one of determining the fin dimensions, the thickness, T, the width, W, and the space between adjacent fins, S, giving a maximum heat transfer for reasonable values of fin weight M and the pressure drop of the cooling air ΔP, but it also involves the determination of these factors when limitations on the fin dimensions are imposed by manufacturing and service requirements. Limitations on the minimum fin thickness and space between fins that can be cast or machined without prohibitive expense may limit seriously the heat transfer for a given pressure drop or fin weight. Furthermore, in applications where the fins must withstand rough handling or where the fins are used for strengthening the cylinder wall, the necessity for having strong fins also may prove to be a limiting factor. Obviously, it is difficult to determine the best fin proportions that will satisfy all requirements.

In this paper it is proposed to take cylinders of one particular diameter, for which considerable data on the pressure drop and the surface heat-transfer coefficient q are available, and show the trends of the several variables for this cylinder. In this manner it is possible to present a clearer picture of what improvements are possible and what penalties are to be paid for these improvements.

Minimum Weight, Pressure Drop, or Power

In most cases the criterion for determining the choice of fin dimensions will be either the weight, the pressure drop, or the power required for cooling. It is desirable to have all these variables as small as possible. However, in almost every case it will be more important to obtain a minimum value for one particular variable. For instance, when the cooling air flow through an engine cowling is induced by the forward movement of the airplane and the slipstream from the propeller, the pressure drop available for forcing the air over the cylinders may be insufficient for cooling at the power output desired. In this case it may be desirable to sacrifice minimum weight in order to obtain sufficient cooling with the limited value of the pressure drop. In cases where the cooling air is supplied by a blower, a wide range of pressures

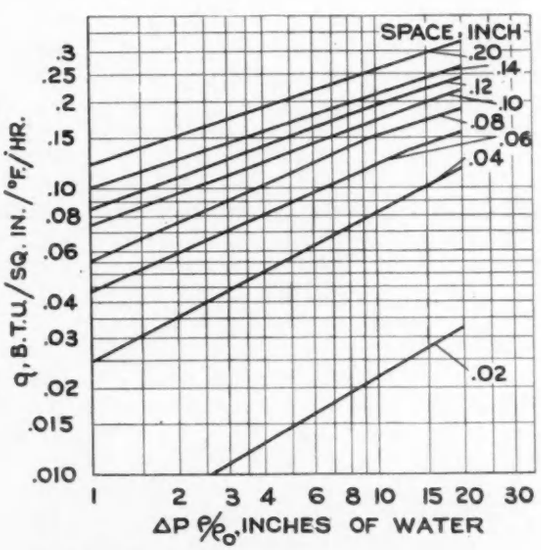

Fig. 1 – Heat-Transfer Coefficient, q, for Various Fin Spaces and Pressure Drops

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 12, 1937.]

[1] See "The Heat-Transfer Processes in Air-Cooled Engines," by Benjamin Pinkel, presented at the Meeting of the Institute of Aeronautical Sciences, New York, N. Y., Jan. 27-29, 1937.

may be available and the power required for cooling may be equally as important as the pressure drop or weight of the fins.

Practical Limits of Fin Dimensions

As will be shown later in the discussion, it is often very desirable to have very thin, closely spaced fins of considerable width. Past experience, however, has shown that it is difficult to machine or to cast fins having these characteristics. For example, steel fins having a width of more than 1 in., a thickness of less than 0.020 in., and a space between fins of less than 0.060 in., may prove to be very difficult and expensive to manufacture. Furthermore, taking the latest cast-aluminum fin designs now on the market as an indicated limit in this field, we find tapered fins having a width of 1.5 in., an average thickness of 0.078 in., and an average space between fins of 0.141 in. These values give only a rough idea of the present limitations on the minimum values of space and thickness. Apparently, if sufficient advantage is to be gained by further decrease in these minimum values, methods for accomplishing that end will present themselves.

The determination of the heat flow from metal fins may be treated as two related problems, one involving the convection of heat from the fin surfaces and the other involving

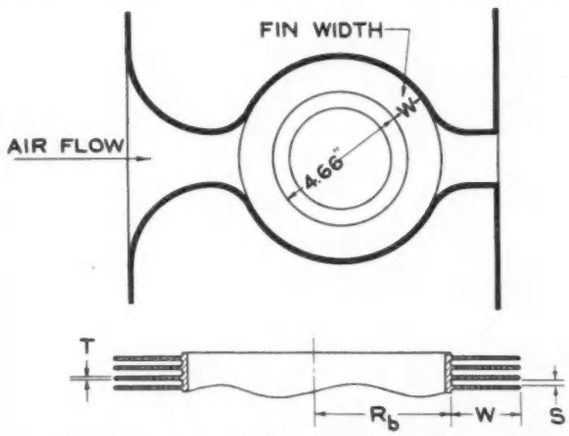

Fig. 2 – Plan of Air Jacket Around Cylinder and Section through Fins

the conduction of heat through the fins to the fin surfaces. The rate at which heat is conveyed from a surface by an air stream is usually expressed as a surface heat-transfer coefficient q. The important factors determining the value of q are the density, velocity, specific heat, viscosity, and thermal conductivity of the air, type of air flow, and fin dimensions. Considerable data are available concerning the effect on q of fin dimensions, jackets, baffles, and various devices for creating turbulence. This phase of the problem is a large subject in itself and need not be considered here other than for determining the relation between the pressure drop across the cylinder and the surface heat-transfer coefficient q for the particular cylinders chosen for this discussion. The curves in Fig. 1 show this relation for a range of fin-space values. These data, which are taken from a report now in the process of publication, are for a cylinder of 4.66-in. diameter enclosed in a jacket as shown in Fig. 2. The cooling air was drawn through the jacket by means of a blower. These and other tests show that q is practically independent of fin dimensions with the exception of the fin space S. The relation between pressure drop and weight-velocity $V\rho g$ is taken from a report of other tests with similar apparatus now in process of publication and is shown in Fig. 3. The pressure-drop measure-

[2] See N.A.C.A. Technical Report No. 488, 1934: "Heat Transfer from Finned Metal Cylinders in an Air Stream," by Arnold E. Biermann and Benjamin Pinkel.

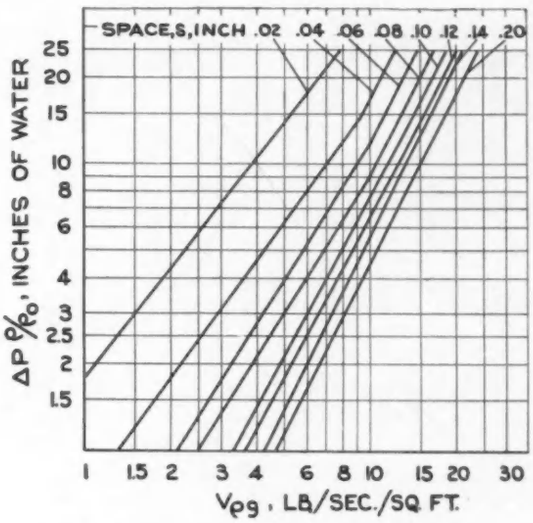

Fig. 3 – Effect of Mass Flow of Cooling Air on Pressure Drop

ments shown in both Figs. 1 and 3 include only the drop across the cylinder.

For the second part of the problem, that involving the conduction of heat through the fins, the following fundamental equation has been used[2]

$$U = \frac{q}{S+T}\left[\frac{2}{a}\left(1+\frac{W}{2R_b}\right)\tanh aW' + S_b\right]$$

where

$$a = \sqrt{\frac{2q}{kT}}$$

U, overall heat-transfer coefficient, B.t.u. per sq. in. of base area per hr., per deg. fahr. temperature difference between the cylinder wall and the entering cooling air.

q, surface heat-transfer coefficient, B.t.u. per sq. in. total surface area per hr., per deg. fahr. temperature difference between the surface and the entering cooling air.

k, thermal conductivity of metal, B.t.u. per sq. in., per deg. fahr. through 1 in. per hr. (2.17 for steel, 7.66 for aluminum Y alloy).

T, average fin thickness, in.

S, average space between fins, in.

W, fin width, in.

W', $W + \frac{T_t}{2}$, effective fin width.

T_t, fin-tip thickness, in.

R_b, radius from center of cylinder to fin root, in.

S_b, distance between adjacent fin surfaces at the fin root, in.

This equation has been verified experimentally for steel fins for a wide range of fin dimensions. In addition, limited tests with copper fins have been compared with the calculated values using this equation. Experiments have shown that this equation holds equally well for tapered or triangular-sectioned fins provided that the average values of fin thickness and space are used in the calculations. The tapered-fin sections give a slightly greater heat transfer for a given weight than the rectangular sections, and are preferred for this reason.

Maximum U for a Given Weight and ΔP

A solution of the foregoing equation for maximum U for a given weight of material or minimum weight for a given U depends upon obtaining a simple mathematical relation between S and q for values of ΔP. A cross-plot of Fig. 1 will show that no simple relation exists between S and q, making a solution of the equation for maximum U practically impossible. Furthermore, in this paper the assumption is made that, in practice, limiting values of the fin thickness and space between fins will be required. This is an added complication making a direct solution by equation more difficult. The

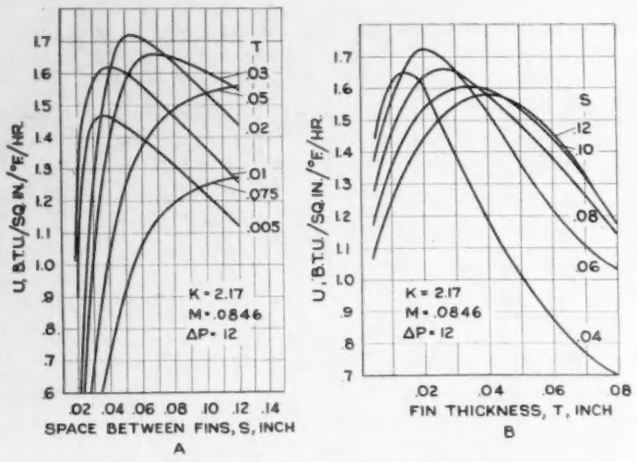

Fig. 4 – Variation of U with S and T for Constant M and ΔP

solution to the problem has been worked out graphically in the following manner:

Two series of graphs for both aluminum and steel were constructed. One series was constructed for a constant amount of fin material in which ΔP was held constant and U was plotted against S for various values of T as shown in the example in Fig. 4*A* (left). The other series was constructed for constant values of ΔP in which U was plotted against T for various values of S as shown in Fig. 4*B* (right). The weight of fin material was calculated from

$$M = c\frac{WT}{S+T}\left(1+\frac{W}{2R_b}\right)$$

where c is the specific weight of the material, lb. per cu. in. (0.282 for steel, 0.101 for aluminum, Y alloy).

An inspection of the two graphs in Fig. 4 shows a peaking point of U on each curve. In Fig. 4*A* it may be noticed that, for a fin thickness of 0.01 in., the heat transfer is a maximum when the fin space is 0.04 in. However, Fig. 4*B* shows that, for a fin space of 0.04 in., the heat transfer is a maximum at 0.015 instead of 0.01 as before. This difference in results depends upon whether S or T is specified and occurs for all peaking points except that for the curves having a maximum peaking value of U. At this point all values are the same for both graphs. These curves, then, show that, if the value of S is specified, the heat transfer and the other dimensions may be different for the same weight of materials than if T had been specified.

From graphs of the type shown in Fig. 4 the final graphs, of which examples for aluminum and steel are shown in Figs. 5 and 6, were plotted from the values corresponding to the peaking points on these curves. Graphs of this type were constructed for a range of fin weights for the conditions in which S is the specified limiting value and for conditions in which T is the limiting value. As this work is too voluminous to include in this paper, only one specified S graph for each metal has been included and they are shown in Figs. 5 and 6.

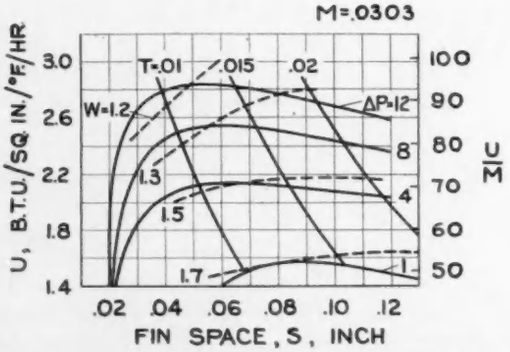

Fig. 5 – Optimum Dimensions for Aluminum Fins when the Fin Space Is Specified

In order to become more fully acquainted with these graphs, let it be required to determine the fin dimensions for steel, giving maximum U for a ΔP of 4 for a fin weight M of 0.0846 lb. per sq. in. of wall area in which the minimum value of T shall be 0.025 in. and S shall not be less than 0.10 in. From the graph of specified T, for these conditions (Fig. 6), it is found that for a T of 0.025 in. the corresponding value of S is smaller than the allowable set as the minimum. Turning to the corresponding graph for specified S (Fig. 6) the solution is $U = 1.26$, $S = 0.10$, $T = 0.032$, $W = 1.03$. If the resulting fin width from this solution is too large to meet requirements, another solution may be obtained by choosing a larger minimum S or possibly by choosing a smaller value of fin weight.

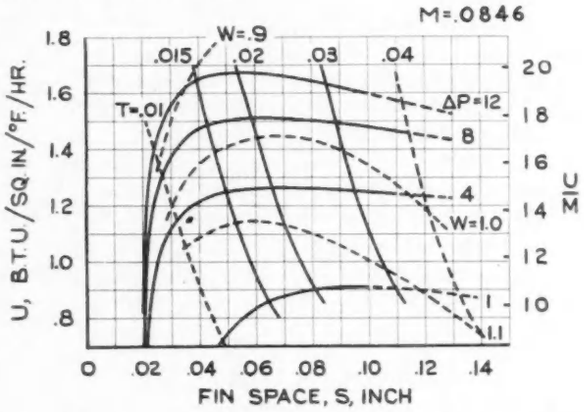

Fig. 6 – Optimum Dimensions for Steel Fins when the Fin Space Is Specified

As the graphs in Figs. 5 and 6 are constructed for a constant weight of fin material, the curves of U also serve for values of the weight criterion U/M. Furthermore, since each curve of U is constructed for a constant ΔP, the variation of the criterion $U/\Delta P$ is at once apparent. From these figures it is therefore seen that the fin dimensions giving maximum heat transfer for a given weight also give maximum heat transfer for a given pressure drop.

Several points about these graphs are of particular interest. The wide range, in most cases, over which both T and S may be varied without much change in U is very noticeable, especially for steel at the low pressure differences. In general, the peaking value of U occurs at larger values of S and T as the pressure drop is reduced. Furthermore, when T is specified, the peaking point is sharper and U falls off more rapidly as T is increased. A comparison of the thickness of the aluminum and steel fins shows that the aluminum fins are much thinner for maximum heat transfer than are the steel fins. This result is unfortunate as very thin, wide aluminum fins are too frail and too difficult to construct to be considered practical. Incidentally, this same objection applies to the use of copper as fin material. Owing to the very high thermal conductivity of copper, the optimum thickness for maximum heat transfer for a given weight or pressure drop is obtained with extremely thin fins. Calculations show that copper fins may compare favorably with aluminum if fins of paper thickness can be made practical.

The fin dimensions corresponding to maximum U for a wide range of fin weights have been plotted against U as shown in Fig. 7 for steel and aluminum. The dimensions of typical fins representing the best modern air-cooled-engine practice are shown for comparison on these curves. The most

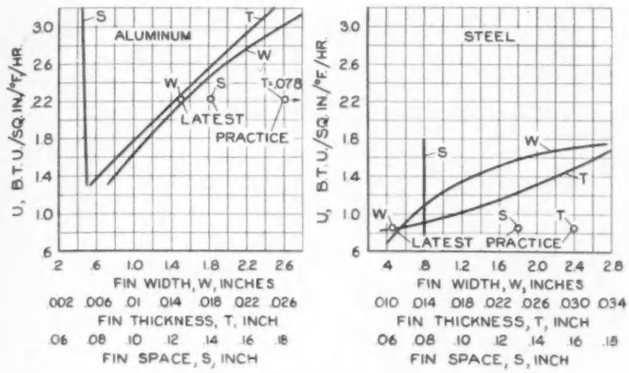

Fig. 7 – Fin Dimensions for Maximum Heat Transfer at a ΔP of 4

noticeable difference between the dimensions representing maximum U and the typical fins is in the much larger S and T values of the latter.

Fig. 8 shows the variation of maximum U with the weight of fins for steel and aluminum. In these curves U was increased by changing fin dimensions according to the values shown in Fig. 7. The fin proportions at each point were the optimum for that heat transfer and weight at a ΔP of 4 in. of water. Obviously, a great many of the fin dimensions that correspond to the values on these curves are impractical from a service or manufacturing standpoint. This criticism is especially true of the aluminum fins for low values of U where the fins are very thin. If a lower pressure drop had been chosen for these curves, all fin dimensions would be increased, making the aluminum fins more practical. As the curves are shown, they indicate a maximum by which other fin designs may be compared. The values of U and M for typical fin designs representing the latest practice for aluminum and steel have been shown on this graph for comparison.

It is noticed that the heat transfer of the fin design representing the latest practice for aluminum can be increased about 23 per cent without increasing the weight, or the weight can be reduced about 50 per cent for the same heat transfer. The possible improvement for steel fins is much less. Although this possible improvement in heat transfer for the same fin weight is not very great, it must be remembered, as previously mentioned, that such an improvement in engine cooling would permit a much greater specific power output than this percentage would indicate, or possibly better fuel economy by permitting the use of higher compression ratios.

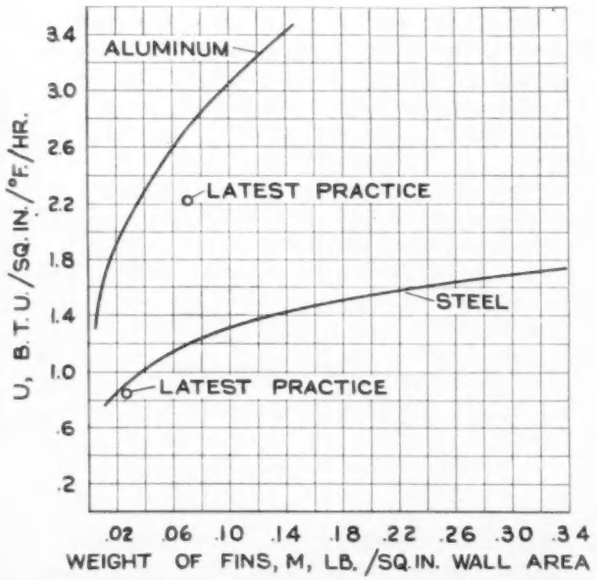

Fig. 8 – Variation of Maximum U with Weight of Fins for a ΔP of 4

The question is often asked as to how much improvement can be obtained by using aluminum fins instead of those of steel. Fig. 8 gives the answer to this question for fins having optimum proportions. For the same fin weight at a ΔP of 4 the aluminum fins transfer more than 2¼ times as much heat as the steel fins. Obviously, this value will be much less for designs in which the fin thickness and space are larger than for the optimum proportions.

The aluminum fins of best proportions are probably too thin for practical use. As a result, it is of interest to determine the effect on U and M of specifying the minimum value of the fin thickness. This information is shown in Fig. 9 for specified thicknesses of 0.030, 0.050, and 0.075 in. These curves show that the reduction in U caused by increasing the thickness over that for the optimum dimensions is less at large values of U and increases gradually as the fin weight and heat transfer are reduced. For a fin weight of 0.071 lb. per sq. in. of wall area, the reduction in U incurred by specifying fin thicknesses of 0.030, 0.050, and 0.075 in. is 2.2, 6.9, and 17.6 per cent respectively. The respective values of S and W for the foregoing conditions corresponding to the optimum fin proportions and the proportions, when a thickness of 0.075 in. is specified, are 0.075 and 0.16 for S and 2.0 and 1.6 for W.

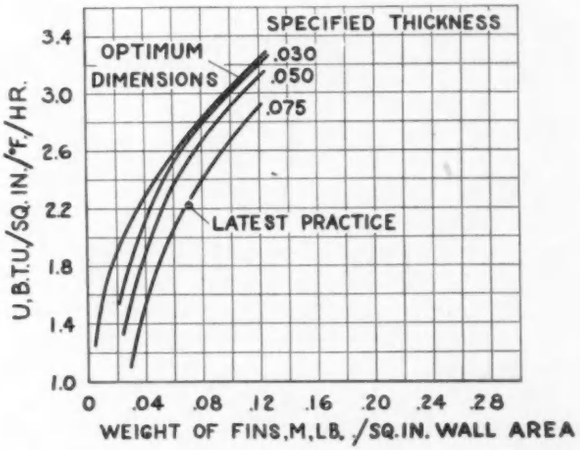

Fig. 9 – Maximum Heat Transfer Obtained with Various Specified Thicknesses for Aluminum with a ΔP of 4

Increasing the Heat Transfer

From Fig. 8 it is apparent that considerable improvement in the heat transfer can be obtained by increasing the fin weight providing fins of the best proportions are used. For instance, with a ΔP of 4, the heat transfer of aluminum fins having a U of 2.2 can be increased 24 per cent by doubling the fin weight. Similarly, for steel the value of U at 0.9 can be increased about 21 per cent by doubling the fin weight. In general, these improvements are obtained by increasing the thickness and width. Although the increase in engine power made possible through improvements on cooling may fully warrant additional weight of fins, it may prove more practical to increase the cooling by resorting to higher air velocities over the fins, even though some form of blower is required. In this case the power required to force the air between fins must be considered.

Power Required for Cooling

Although no exact determinations have been made of the best fin proportions for maximum heat transfer with a minimum expenditure of power, there are indications to show that the fin designs giving maximum heat transfer for a given

pressure drop are almost the same as those giving maximum U for a given power. This result is illustrated by tests in which it was found that maximum U occurs at practically the same fin space for given values of the velocity between fins as for given values of the power required for cooling. The variation of maximum U with respect to the power required for cooling with values of ΔP of 4 and 12 is shown in Fig. 10. In this figure the curves for aluminum and steel with a ΔP of 4 show that very little reduction in power is possible by changing from the fin design representing the latest practice to the fin designs for maximum heat transfer for a given pressure drop. However, for a ΔP of 12, the conventional fin design for aluminum requires 33 per cent more power than the design for maximum heat transfer. The corresponding value for steel is 38 per cent.

The cost in cooling power caused by increasing ΔP from 4 to 12 may be compared with the reduction in fin weight permitted for the same U in both cases. Thus, for a U of 3.0, an increase in ΔP from 4 to 12 causes an increase in the power required for cooling of 195 per cent and a reduction in fin weight of 60 per cent. This example illustrates the fact that increasing the cooling by increasing the pressure drop requires considerable increase in the power. In the preceding illustration it is necessary to evaluate fin weight in terms of pressure drop and the power required for cooling. Obviously,

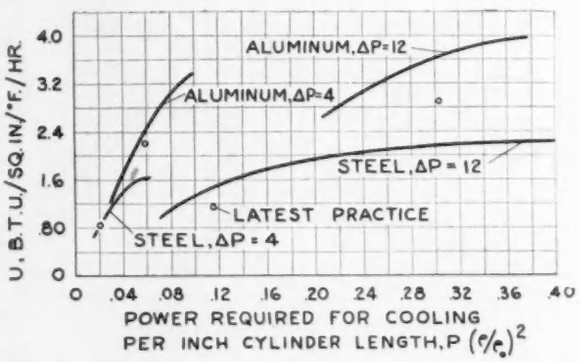

Fig. 10 – Variation of Maximum U with Power Required for Cooling

this evaluation is not easy as several practical problems will have to be considered.

Application of Results

The results of this analysis are strictly applicable only to the cylinder and test conditions set forth. Any factor such as cylinder diameter, baffles, or turbulence devices that changes the effective surface heat-transfer coefficient q for a given ΔP will change these results according to the amount that ΔP is changed. As there is little change in optimum fin dimensions with a variation of ΔP from 4 to 12 it is believed that the results of an analysis of this kind are sufficiently accurate to indicate trends for a wide range of conditions.

Conclusions

(1) This analysis shows that a considerable improvement in the heat transfer of conventional aluminum fin designs is possible by correctly proportioning the fin dimensions.

(2) The optimum thickness of fins decreases as the thermal conductivity of the material increases.

(3) Correctly proportioned aluminum fins will transfer more than 2¼ times as much heat as steel fins for the same weight and pressure drop.

(4) The best fin proportions for maximum heat transfer for a given pressure drop are also good for obtaining a high heat transfer for a given power expenditure.

Oil Filters in Public Utility Fleet Operation

(*Continued from page 387*)

Thus, viewed in this light, it appears that we should make every effort to obtain the optimum life from oils. This objective, I believe you will agree, necessitates the invention of some practical test that will determine accurately the life of an oil. Personally, it is my belief that oil can be used indefinitely, provided it is either continuously purified while in service or removed periodically from service and purified.

Furthermore, it seems logical to me that more efficient results should be secured by the continuous removal of impurities rather than permitting them to accumulate to perhaps a dangerous extent and then removing them by a batch treatment. Obviously, when batch treatment is practised, the oil is only relatively clean for a short period of time and, consequently, is rendering service most of the time while heavily loaded with harmful impurities. Continuous purification, if effective, insures that the oil is comparatively clean at all times.

The purification of oil in the powerplant field is an accepted necessity for economical operation and has been for many years. I have never been able to understand fully the logic of some large utility firms who scrupulously purify all of their steam-turbine oils and, at the same time, pour their crankcase drainings to the sewer. Certainly, if reclaimed oil is good enough for a $200,000 or $300,000 machine upon which the whole community depends for light and power, it should be good enough for a bus upon which a few people depend for transportation to and from the office, tea parties, bridge parties, and so on.

There is no question in my mind but that a good filter, correctly installed and properly serviced, will return a handsome profit. The servicing is very important. All filters on the market today require some servicing. Some require more attention than others, and some are easier and more convenient to service but, nevertheless, some attention must be given to them all. For best results the servicing of a filter should be scheduled and provided for during the regular inspection period.

General Conclusion

It cannot be said that an oil filter is essential to the operation of a motor vehicle; but the use of one of the more efficient types will, in my opinion, greatly extend the interval between oil changes, and the saving in oil thereby made, in general, will more than cover the cost of the filter. Furthermore, the use of a filter will generally reduce piston, ring, and bore wear and valve and ring-sticking and, in this way, reduce oil consumption and possibly fuel consumption. Some of the present filters are capable of maintaining the acidity of the oil at a low value, thereby minimizing corrosive effects, especially on the newer, copper-lead and cadmium-silver bearing alloys.

Acknowledgment

I wish to take this opportunity to express my appreciation to the AC Spark Plug Co., The DeLuxe Products Corp., Fleming Mfg. Co., Handy Governor Corp., Motor Improvements, Inc., Olixir Products Co., Rega Mfg. Co., W.G.B. Oil Clarifier, Inc., Autocar Co., Diamond T Motor Car Co., General Motors Truck and Coach, International Harvester Co., International-Plainfield Motor Co., The Studebaker Corp., Waukesha Motor Co., *Fleet Owner*, and *Bus Transportation* for furnishing valuable information and illustrations without which this paper could not have been prepared.

The American Picture – Diesel Fuel Research

By C. G. A. Rosen
In Charge of Diesel Development, Caterpillar Tractor Co.

PRIMARILY intended as a discussion and amplification of the paper of Messrs. Boerlage and Broeze presented at the April, 1936, meeting of the American Chemical Society[1], this paper reviews fuel research conducted at the San Leandro laboratory of the Caterpillar Tractor Co. and, therefore, is limited to the precombustion-chamber type of Diesel engine burning California-base fuels.

The paper describes investigations of ignition quality, fuel-spray characteristics, and injection phenomena by means of a single-cylinder test unit fitted with a quartz observation window, stroboscope, timing disc and phase-changing device.

A discussion of the products of incomplete combustion as influenced by compounded lubricants and ring-belt temperatures concludes the paper.

THIS paper was intended primarily to discuss the paper of Messrs. Boerlage and Broeze, presented at the April, 1936, meeting of the American Chemical Society[1], in the light of fuel research conducted at the San Leandro laboratory of the Caterpillar Tractor Co. This discussion, therefore, will be limited explicitly to results obtained in the precombustion-chamber type of Diesel engine burning California-base fuels.

Perhaps this course of action is justified in that a third type of combustion system is brought into the picture, thus amplifying the scope of Boerlage and Broeze's paper and, furthermore, the local interest in Western-produced fuels should place emphasis on this type of product in a meeting of this kind. The limited time available also dictates brevity of subject matter.

The major premise of the aforementioned paper may be briefly stated as follows:

Maximum efficiency in the Diesel cycle is mainly dependent upon two essential factors:

(1) Engine conditions as they affect mixture formation such as injection characteristics, swirl or turbulence, excess of air, wall temperatures, and so on.

(2) Fuel properties, such as: (a) viscosity, (b) ignition quality, and (c) volatility.

The application of these factors to precombustion-chamber functioning may be of interest. The precombustion-chamber type of engine is sufficiently well known, I take it, to eliminate its explanation here. As a Diesel engine, it may be classified as providing an auxiliary combustion-chamber containing perhaps 30 per cent of turbulent air of combustion into which fuel is sprayed at relatively low pressures. Ignition and partial combustion are initiated and the unburned fuel is cracked into a gas which passes into the main combustion-chamber at high velocity for the completion of combustion in the presence of excess air. Fig. 1 shows a cross-section of the Caterpillar Diesel.

A single-cylinder 5¼ x 8-in. laboratory unit of this type was chosen as the experimental engine for these trials. This engine operated normally at 850 r.p.m. and developed 18.5 b.hp. at 100 lb. per sq. in. b.m.e.p. when operating all of its dependent accessories. This unit also was fitted with a quartz window permitting observation of the flame within the main combustion-chamber, and was provided with a timing disc and phase-changing device in conjunction with a stroboscope for determining flame duration. This apparatus is illustrated in Fig. 2.

For ignition-quality determinations Caterpillar Tractor Co. has developed the throttling method as applied to a single-cylinder engine, shown in Fig. 3.

The engine is started and run at no-load at 900 r.p.m. When the cooling-water temperature reaches 70 deg. fahr., the intake-air passage is throttled by means of a calibrated butterfly valve until the engine shows that it is missing as revealed by white smoke in the exhaust. At this point the reading of the throttle angle is taken and converted to cetane number. In checking our results with those obtained by the Volunteer Group for Compression-Ignition Research of the Society, very close correlation was found to prevail.

For the purpose of investigating fuel-spray characteristics and injection phenomena, the apparatus shown in Fig. 4 was developed. This machine can be operated at selected camshaft speeds and the character of the spray observed in the look-box in the open atmosphere. By means of synchronized timing of a neon lamp in relation to the injection position of the fuel-pump plunger, the entire injection period can be studied visually. After obtaining the pertinent characteristics of the spray in the open atmosphere, correlated data are then obtained under cylinder compression pressures.

Having described the apparatus employed for this investigation, specific data were selected to discuss the influence of mixture formation and fuel properties on precombustion-

[This paper was presented at Northern California Section Meeting of the Society, San Francisco, Calif., Sept. 15, 1936.]

[1] See *Industrial and Engineering Chemistry*, October, 1936, pp. 1229-1234; "Combustion Qualities of Diesel Fuel," by G. D. Boerlage and J. J. Broeze.

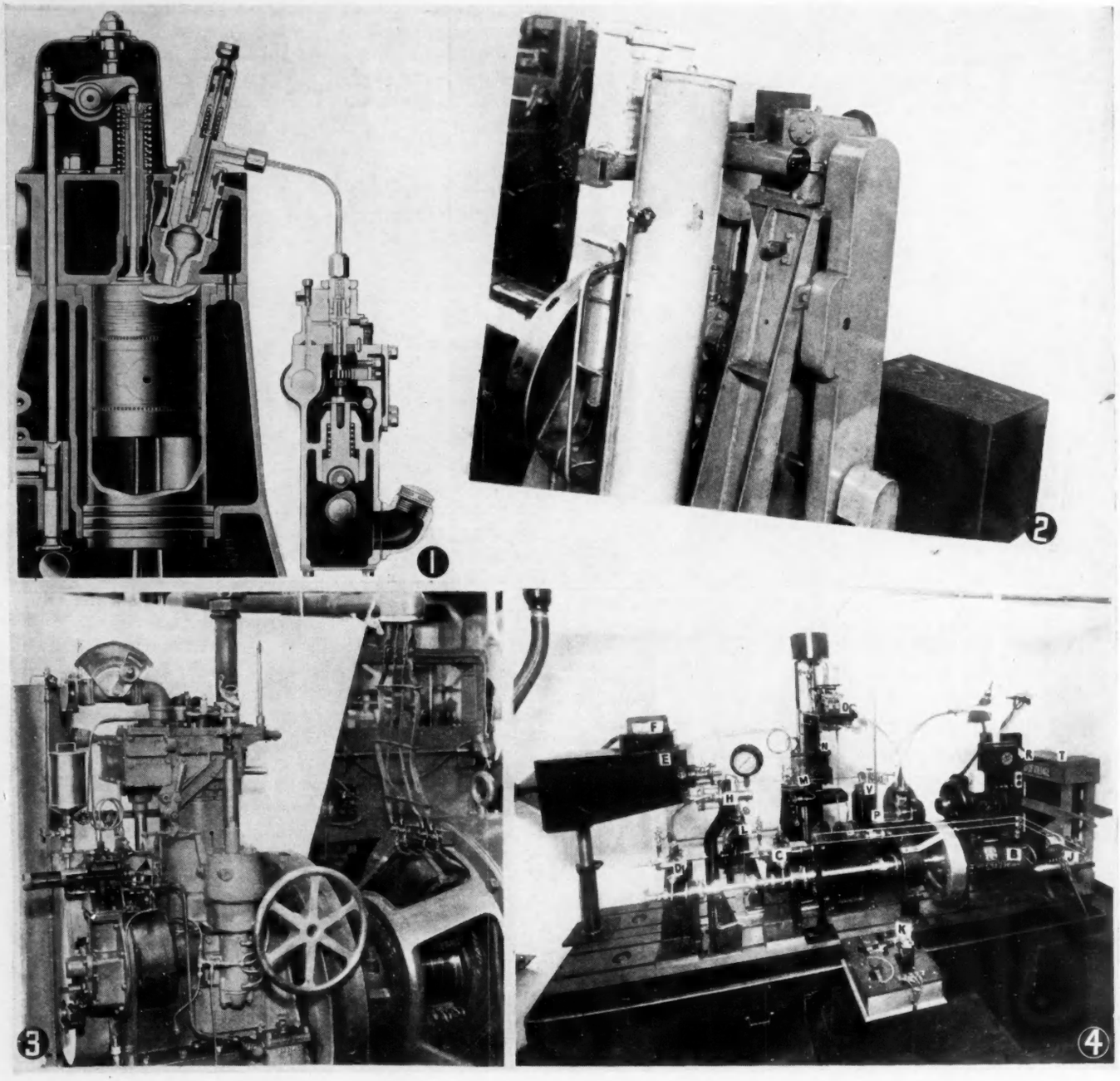

Fig. 1 – Cross-Section of Caterpillar Precombustion-Chamber, Diesel Engine

Fig. 2 – Single-Cylinder Test Engine Fitted with Quartz Window, Timing Disc, Synchronizing Device, Telescope, and Stroboscope

Fig. 3 – Single-Cylinder Test Engine Arranged for Checking Ignition Quality of Fuels by the Throttling Method

Fig. 4 – Spray Machine Used for Determining Fuel-Injection Characteristics and Spray Formations

chamber functioning. Injection characteristics have an important bearing on engine performance.

Fig. 5 illustrates four spray patterns taken with widely varying fuels: The upper pattern was taken with a fuel of 36.5 sec. S. U. viscosity at 100 deg. fahr.; the next lower pattern was taken using a fuel of 42.5 sec. S. U. viscosity at 100 deg. fahr.; the two lower patterns were obtained when using a No. 4 furnace oil of 590 sec. S. U. viscosity at 100 deg. fahr.

The two upper illustrations can be considered representative of California fuels commercially available within the range of overhead distillates.

The two lower diagrams represent conditions obtained when using fuels containing residuals. The two upper diagrams are practically identical: Injection commences at the fuel valve at exactly 6¼ deg. before top-center, in both cases, and terminates injection at almost the same point; the point of beginning of injection is positive and the spray ceases with sharp cutoff without any evidence of secondary injection or dribbling.

In the case of the residual fuels the beginning of injection is considerably delayed and is not constant for every cycle. It is evident that the beginning of injection may vary between cycles as much as 4 deg. The termination of injection is not positive and dribbling prolongs the introduction of fuel far down the combustion stroke. It is to be expected that this fuel would cause a coking or carbonizing of nozzle orifices with continued operation.

For maximum efficiency, combustion-chambers are designed to satisfy fuel-spray characteristics. Nozzle characteristics are

influenced by fuel viscosity. Fig. 6 shows photographs of sprays obtained at exactly the same point in the injection period on two widely different viscosities. The upper photograph illustrates the wide bulbous spray, obtained from a 35-sec. fuel. The dispersion of this spray fills the maximum angle allotted by the combustion-chamber and further illustrates the densified atmosphere at points where the very finely divided fuel particles have lost their energy.

The lower photograph indicates the greater penetrating ability of heavier viscosity fuels (100 sec. in this case), but reveals the narrowness of the dispersion cone. These photographs indicate the influence of fuel viscosities on atomization and distribution.

Low-viscosity fuels permit of fine atomization with broad dispersion but shortened penetration. Higher-viscosity fuels offer greater resistance to atomization causing the larger fuel particles to have greater penetration with corresponding loss of dispersion. The spray character naturally influences evaporation of the fuel droplets. For correct mixture formation the air of combustion must be brought into intimate contact with all fuel particles to prevent the unnecessary wandering of the fuel droplets searching for their equivalent of oxygen. This requirement calls for careful control of air currents to obtain the most homogeneous fuel-air mixture.

In the precombustion-chamber type of Diesel the high velocity of air is directed axially at the core of the spray cone with no deleterious influences arising from fuel particles striking the envelope of the highly heated and insulated precombustion chamber. After initial ignition, further flow of the air of compression into the fuel-spray cone permits of rapid oxidation and gasification of the injected fuel. Spray characteristics are definitely interrelated with turbulence.

The most desirable results are obtained by using a composite fuel spray providing a central core of high penetrating quality counteracting the shredding action of high velocity air currents and an outer skirt or fringe of very finely divided

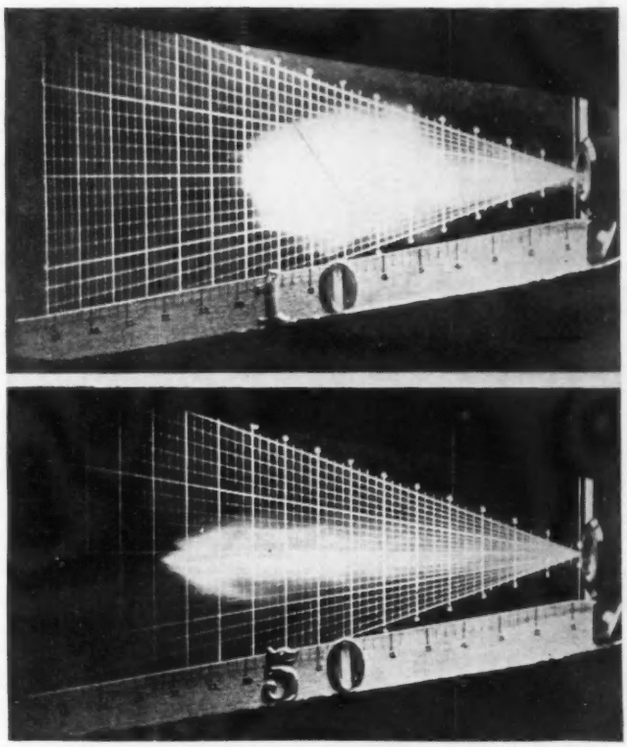

Fig. 6 – Spray Photographs of 35-Sec. S.U. (*above*) and 100-Sec. S.U. (*below*) Fuels at 100 Deg. Fahr.

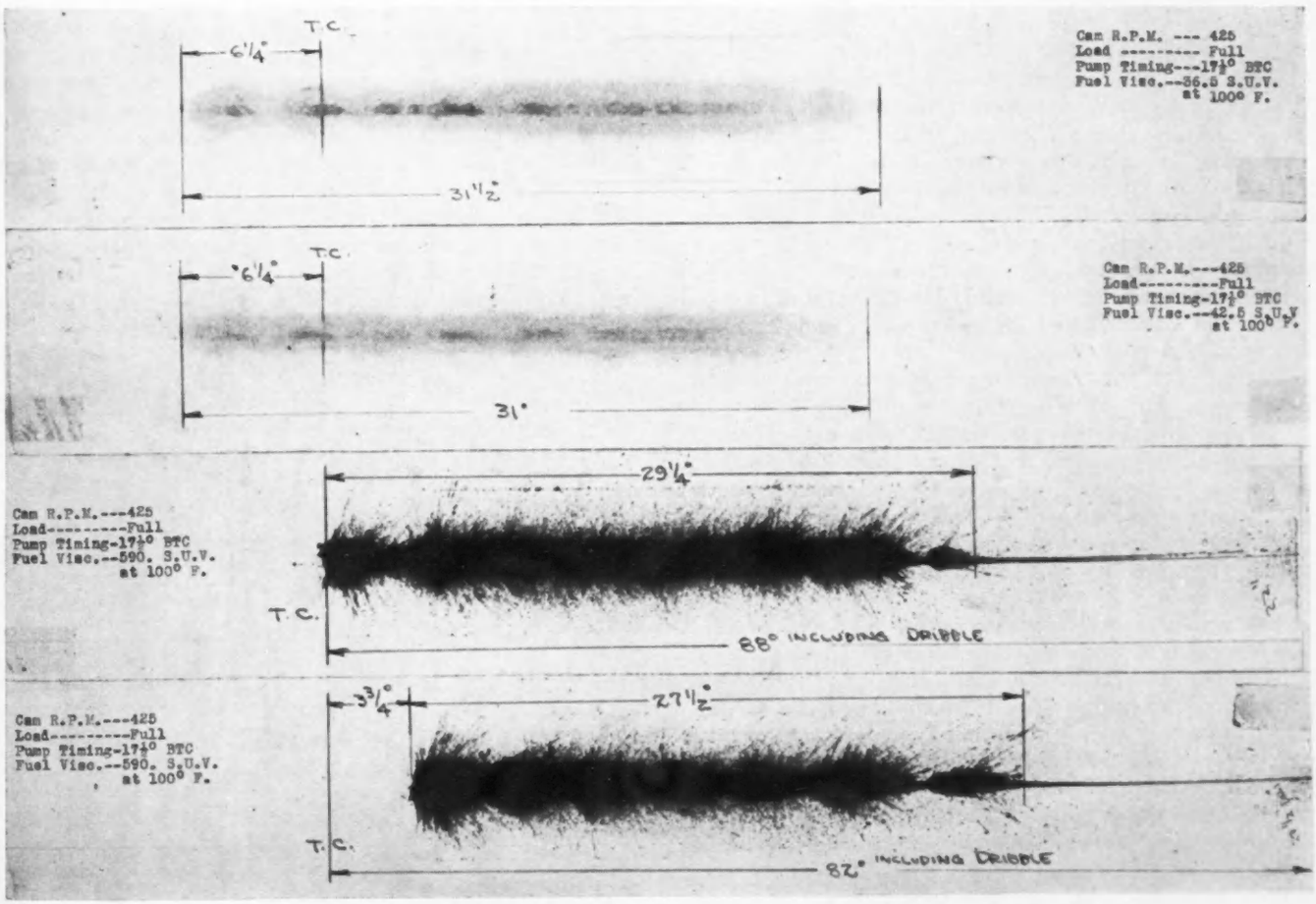

Fig. 5 – Injection Timing and Spray Patterns Obtained from Blotter Machine

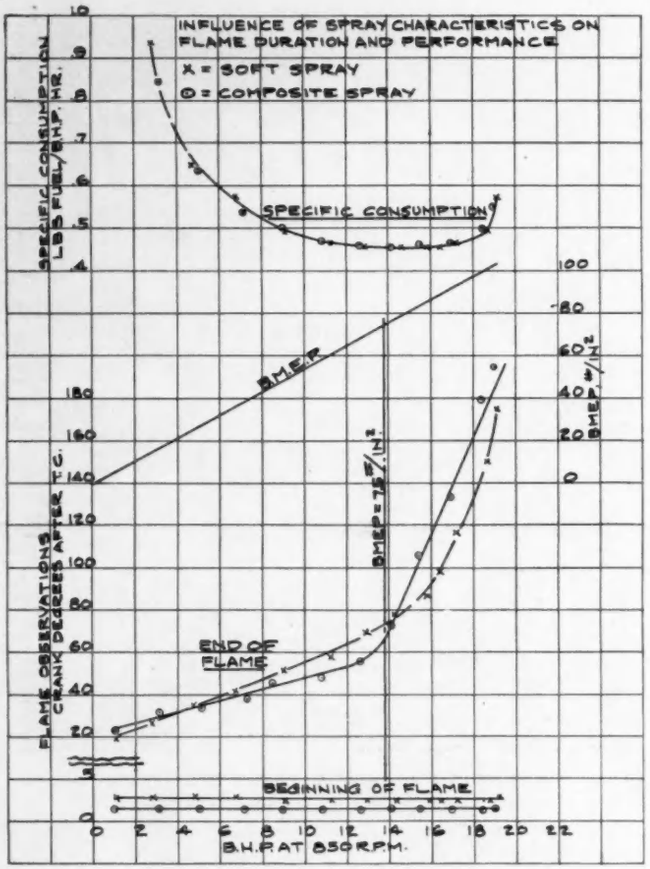
INFLUENCE OF SPRAY CHARACTERISTICS ON FLAME DURATION AND PERFORMANCE
X = SOFT SPRAY
O = COMPOSITE SPRAY
SPECIFIC CONSUMPTION
SPECIFIC CONSUMPTION LBS FUEL/B.H.P. HR.
B.M.E.P.
BMEP = 75 #/IN.²
FLAME OBSERVATIONS CRANK DEGREES AFTER T.C.
END OF FLAME
BEGINNING OF FLAME
B.H.P AT 850 R.P.M.

Fig. 7 – Influence of Spray Characteristics on Flame Duration and Performance

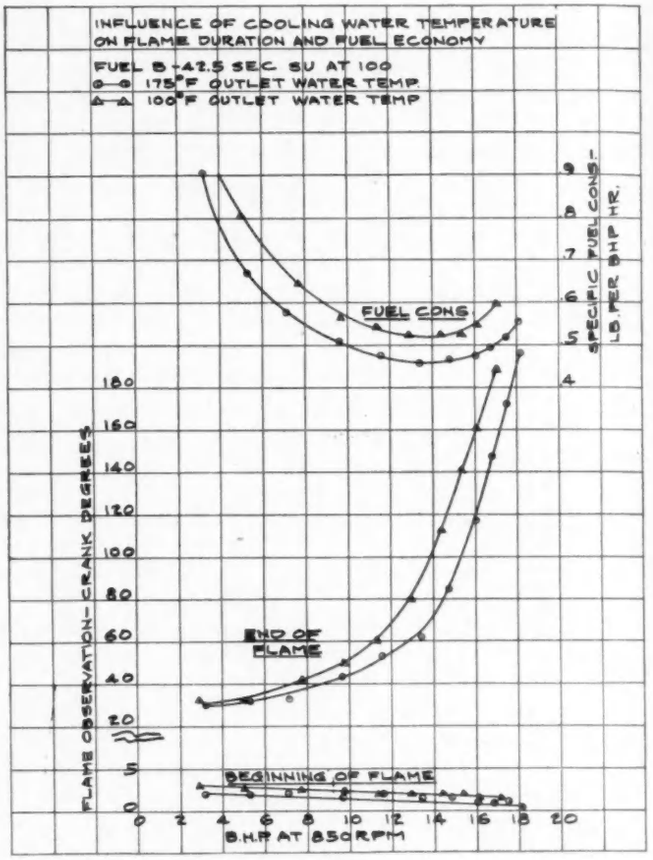
INFLUENCE OF COOLING WATER TEMPERATURE ON FLAME DURATION AND FUEL ECONOMY
FUEL B-42.5 SEC. SU AT 100
O—O 175°F OUTLET WATER TEMP
Δ—Δ 100°F OUTLET WATER TEMP
FUEL CONS.
SPECIFIC FUEL CONS. LB PER B.H.P. HR.
FLAME OBSERVATION—CRANK DEGREES
END OF FLAME
BEGINNING OF FLAME
B.H.P AT 850 R.P.M.

Fig. 8 – Influence of Cooling-Water Temperature on Flame Duration and Fuel Economy

fuel particles to provide short ignition delay by the rapid heating and igniting of extremely small fuel droplets. This type of spray is most satisfactory for engines required to be operated at widely varying speeds. The nucleus of ignition is provided in the outer skirt of finely divided fuel particles and the prevention of over-rich patches of fuel droplets by the highly penetrating character and the slower evaporating qualities of the larger fuel particles in the main core of the spray.

The gasification of the main bulk of the fuel, therefore, is obtained in the precombustion-chamber over wide speed ranges. The gasified fuel traveling into the main combustion-chamber at extremely high velocities quickly finds its equivalent to oxygen by gas dispersion.

The type of spray has its influence on the flame characteristics in the main combustion-chamber. In Fig. 7 the flame duration within the main combustion-chamber is outlined for two types of spray valves. The curves with points marked *X* are produced when using a fuel valve developing a soft spray; the curves with points marked *O* are produced by a fuel valve developing a composite spray. The two fuel valves were developed to their best economy at 850 r.p.m. as indicated by the fuel-consumption curve, yet the composite spray provided greater advantages when operating over wide ranges of speed. It is evident from these curves that the composite spray provided an earlier beginning of flame and, up to about 80 lb. per sq. in. b.m.e.p., showed shorter flame duration. It is rather interesting to note that the fuel finds its equivalent of oxygen for combustion almost as a straight-line function up to about 75 lb. per sq. in. b.m.e.p. Beyond this point, the increased quantity of fuel injection must wander further down the stroke to satisfy its affinity for oxygen, thus increasing flame duration.

Wall temperatures have a considerable influence on the rate of evaporation of the fuel particles. To quote Boerlage:

"As a rule the design of an engine that allows of the highest wall temperatures without undue ill effects on the other part of the process is the most favorable."

This effect is shown clearly in Fig. 8. These curves compare flame duration and fuel consumption at two water-jacket temperatures, namely, 100 deg. fahr. and 175 deg. fahr. The curves quite obviously point to the advantage of the higher water-jacket temperature. At 175 deg. fahr. water-outlet temperature, the beginning of the appearance of the flame in the main combustion-chamber is at least a degree earlier and the flame duration is 40 deg. shorter at full-load operation. Fuel consumption also is improved by over 11 per cent.

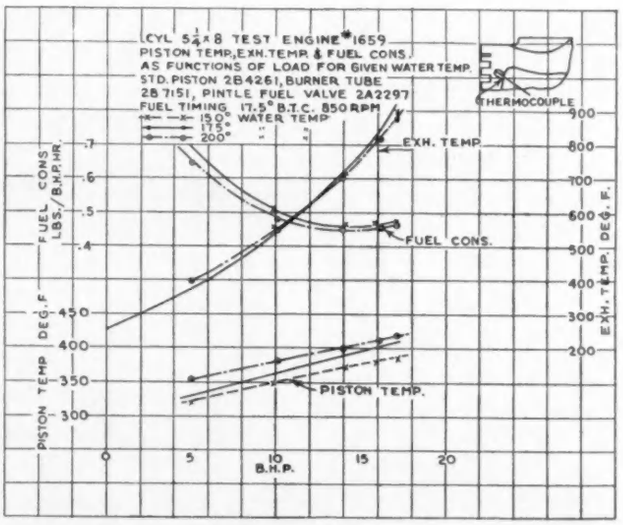
CYL 5½ x 8 TEST ENGINE #1659
PISTON TEMP., EXH. TEMP. & FUEL CONS. AS FUNCTIONS OF LOAD FOR GIVEN WATER TEMP.
STD. PISTON 2B4261, BURNER TUBE 2B7151, PINTLE FUEL VALVE 2A2297
FUEL TIMING 17.5° B.T.C. 850 RPM
150° WATER TEMP
175° " "
200° " "
THERMOCOUPLE
EXH. TEMP.
FUEL CONS.
PISTON TEMP.
FUEL CONS. LBS./B.H.P.HR.
PISTON TEMP. DEG. F.
EXH. TEMP. DEG. F.
B.H.P.

Fig. 9 – Influence of Water-Temperature on Ring-Belt Temperature

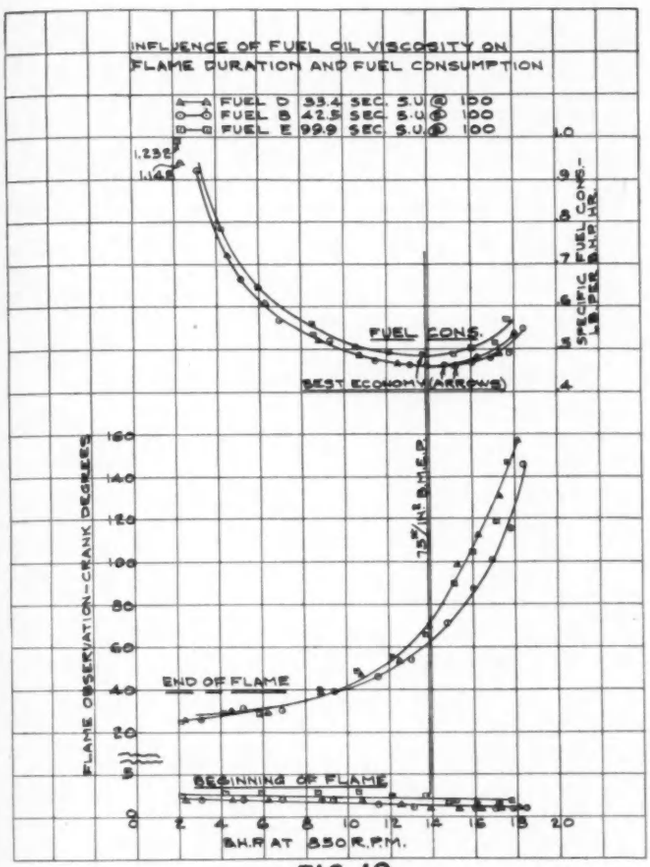
INFLUENCE OF FUEL OIL VISCOSITY ON FLAME DURATION AND FUEL CONSUMPTION
FUEL D 33.4 SEC. S.U. @ 100
FUEL B 42.5 SEC. S.U. @ 100
FUEL E 99.9 SEC. S.U. @ 100
FUEL CONS.
BEST ECONOMY (ARROWS)
SPECIFIC FUEL CONS. LB. PER B.H.P. HR.
FLAME OBSERVATION - CRANK DEGREES
75% M.E.P.
END OF FLAME
BEGINNING OF FLAME
B.H.P. AT 850 R.P.M.

FIG. 10

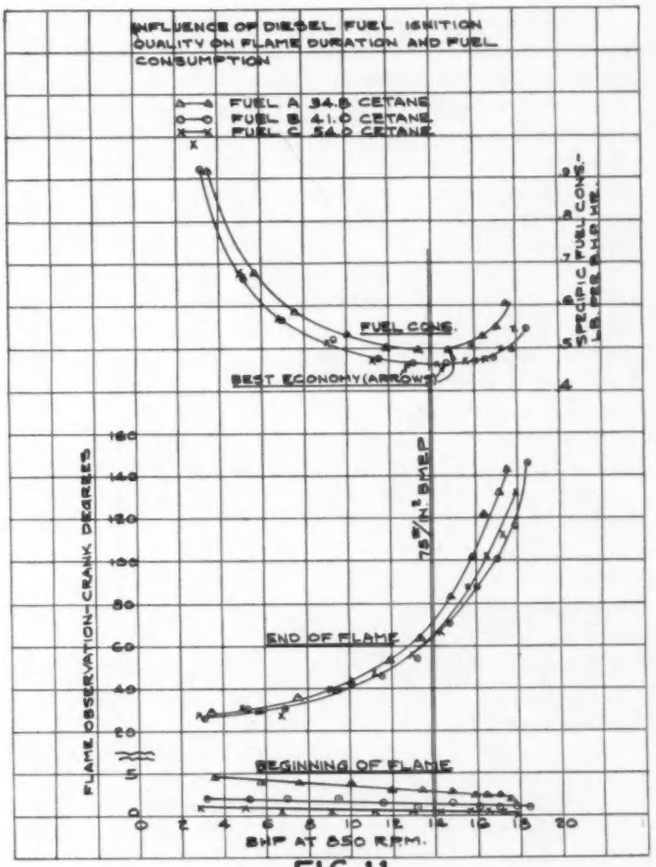
INFLUENCE OF DIESEL FUEL IGNITION QUALITY ON FLAME DURATION AND FUEL CONSUMPTION
FUEL A 34.8 CETANE
FUEL B 41.0 CETANE
FUEL C 54.0 CETANE
FUEL CONS.
BEST ECONOMY (ARROWS)
SPECIFIC FUEL CONS. LB. PER B.H.P. HR.
FLAME OBSERVATION - CRANK DEGREES
75% M.E.P.
END OF FLAME
BEGINNING OF FLAME
B.H.P. AT 850 R.P.M.

FIG. 11

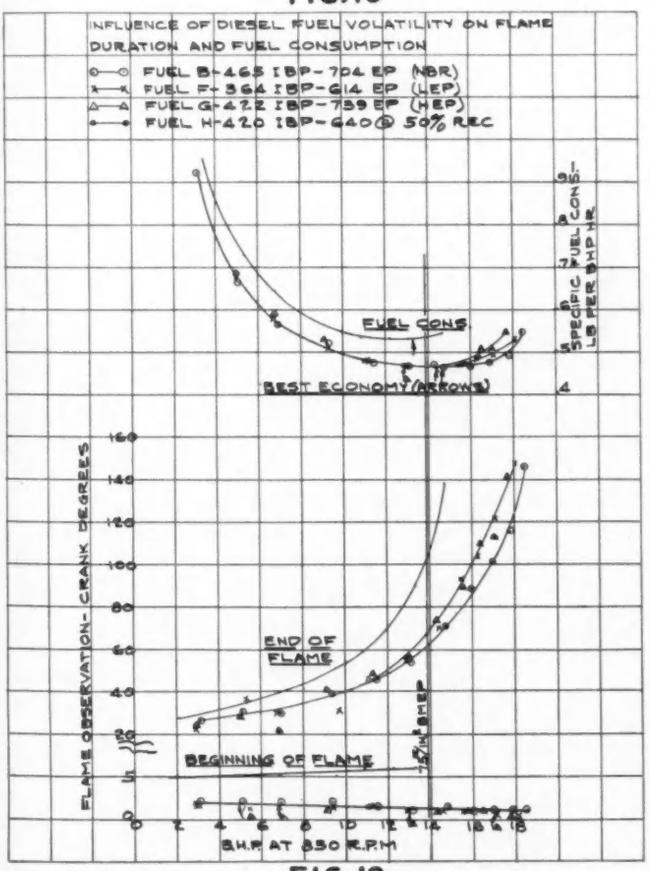
INFLUENCE OF DIESEL FUEL VOLATILITY ON FLAME DURATION AND FUEL CONSUMPTION
FUEL B-465 IBP-704 EP (NBR)
FUEL F-364 IBP-614 EP (LEP)
FUEL G-422 IBP-739 EP (HEP)
FUEL H-420 IBP-640 @ 50% REC
FUEL CONS.
BEST ECONOMY (ARROWS)
SPECIFIC FUEL CONS. LB. PER B.H.P. HR.
FLAME OBSERVATION - CRANK DEGREES
END OF FLAME
75% M.E.P.
BEGINNING OF FLAME
B.H.P. AT 850 R.P.M.

FIG. 12

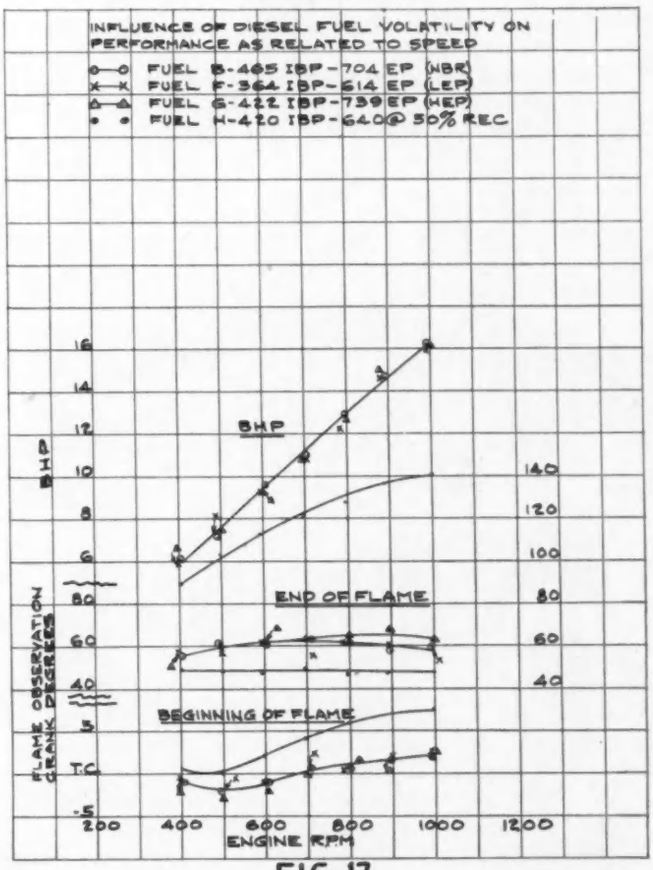
INFLUENCE OF DIESEL FUEL VOLATILITY ON PERFORMANCE AS RELATED TO SPEED
FUEL B-465 IBP-704 EP (NBR)
FUEL F-364 IBP-614 EP (LEP)
FUEL G-422 IBP-739 EP (HEP)
FUEL H-420 IBP-640 @ 50% REC
B.H.P.
END OF FLAME
BEGINNING OF FLAME
T.C.
FLAME OBSERVATION CRANK DEGREES
ENGINE R.P.M.

FIG. 13

Fig. 10 – Influence of Fuel Viscosity on Flame Duration and Fuel Consumption

Fig. 11 – Influence of Diesel Fuel Ignition Quality on Flame Duration and Fuel Consumption

Fig. 12 – Influence of Diesel Fuel Volatility on Flame Duration and Fuel Consumption

Fig. 13 – Influence of Diesel Fuel Volatility on Performance as Related to Speed

The influence of jacket-water temperature upon the envelope surrounding the combustion-chamber can be appreciated readily by referring to Fig. 9. Here the influence of jacket-water temperature is shown on the ring-belt temperature immediately behind the top piston-ring. These temperatures are well within the normal range of operation of a tractor Diesel engine, from 150 deg. fahr. to 200 deg. fahr. The average piston temperature behind the top ring at full load varies from 380 deg. fahr. to 415 deg. fahr. The temperature of the envelope surrounding the burning gases is, therefore, of vital influence in the rapid completion of combustion and the resulting economies in fuel consumption.

The influence of fuel properties on precombustion-chamber Diesel functioning, involves the factors of viscosity, fuel ignition quality, and volatility.

The study of flame duration is given particular emphasis in view of the importance attributed to the fourth phase of combustion as expressed by A. G. Marshall's analysis[2] of Boerlage and Broeze's paper.[1] Phase 4 refers to the combustion period after the end of injection when an inevitable amount of after-burning takes place in every engine. The better the mixture, the less will be the extent of this after-burning. If the mixture is poor, the combustion will not be completed. The various fuel factors, therefore, are discussed in this paper in relation to flame duration and fuel consumption.

Three fuels are selected for study in Fig. 10, varying from 33.4-sec. viscosity to 99.9-sec. viscosity. The normal range of overhead distillates as represented by the 33.4-sec. fuel and the 42.5-sec. fuel shows the same beginning of flame and practically the same fuel-consumption characteristics. The higher-viscosity fuel shows a later beginning of flame and a longer duration of flame at the higher loads. The fuel consumption also is correspondingly greater.

The ignition quality bears a direct influence on the beginning of flame as illustrated in Fig. 11. This influence is to be expected logically in view of the shorter ignition lag provided by the higher-cetane fuels. The flame duration is greatest with the fuel of lowest cetane number. Fuel consumption is distinctly better over the entire load range with fuels above 40-cetane ignition quality.

[2] Also presented at the Northern California Section Meeting of the Society, San Francisco, Calif., Sept. 15, 1936.

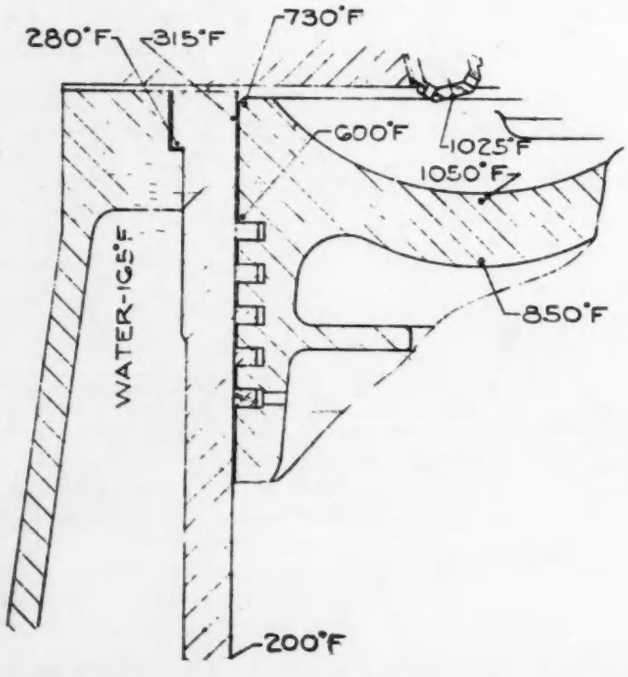

Fig. 14 – Temperature Variation of Cardinal Points in the Combustion-Chamber Envelope

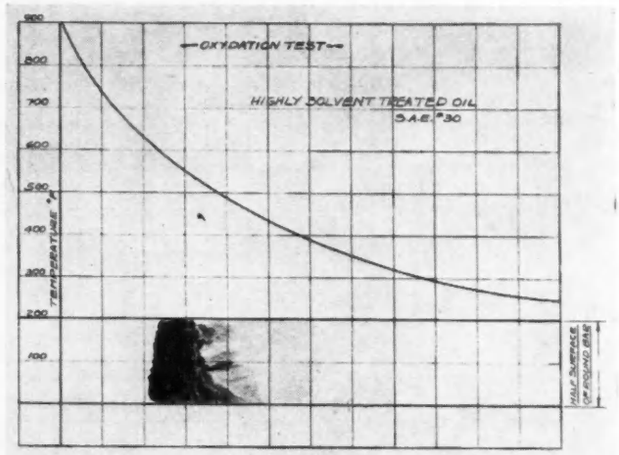

Fig. 15 – Products Obtained in Oxidation Test of Highly Solvent-Treated Lubricating Oil

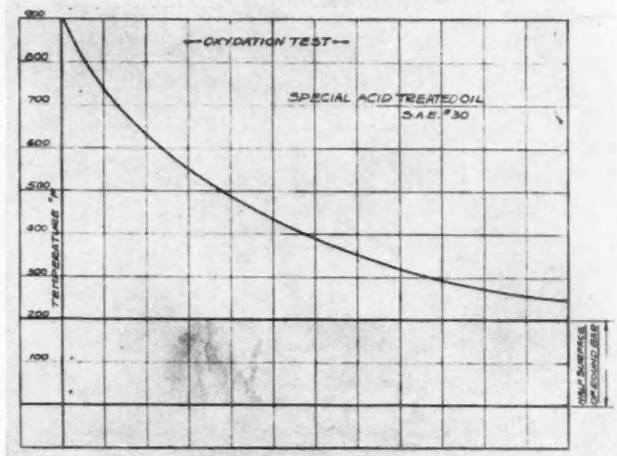

Fig.16 – Products Obtained in Oxidation Test of a Special Acid-Treated Lubricating Oil

No direct comparison can be made with Boerlage's data on volatility, but considerable interest has been manifested in this country as to the influence of end point and boiling range on engine performance. Those fuels normally found within the range of overhead distillates, such as *B*, *F*, and *G*, Fig. 12, exhibit comparable flame-duration characteristics. Fuel *H*, which permitted only 50 per cent recovery at 640 deg. fahr., shows decidedly different flame characteristics. The beginning of the appearance of flame is quite late, and the end of flame is carried a considerable distance down the expansion stroke.

Fuel *H* represents a No. 4 furnace oil containing residuals, and is the type which produced the delayed spray pattern shown in Fig. 5. The fixed-rack setting of the fuel pump permitting comparable power development for Diesel distillates, would not permit of an equivalent rating with the No. 4 furnace oil due to the higher fuel consumption and the lower pump-filling efficiency of this higher viscosity fuel.

Fig. 13 illustrates the influence of boiling range on performance as related to speed. The beginning of flame for the residual fuel (Fuel *H*) becomes considerably later as the speed increases when referred to the overhead distillates. The end of flame is practically a straight line for the fixed-rack setting of the fuel pump, producing a much lower brake horsepower output than obtained with the overhead distillates.

In writing a fuel specification for precombustion-chamber Diesel engines operating at speeds up to 1500 r.p.m., the foregoing data would dictate the following factors:

Type – a completely distilled mineral oil.

Viscosity – 35 sec. S. U., minimum.

Ignition Quality – 40 cetane minimum with preference for the higher values.

Volatility – acceptable within the range of overhead distillates.

Discussion of Products of Incomplete Combustion. – Boerlage and Broeze's paper[1] goes into considerable detail on chemical considerations in after-burning and unburned products. They state:

"As to the reactions during the flame period, although it has not been shown that on the whole their velocity is so high as not to impede the process, they are still of great interest from the point of view of the products of incomplete combustion due to different kinds of after-burning."

They also call attention to the three typical products of incomplete combustion:

(1) Aldehydes and varnishes produced by chilled hydroxylation.

(2) Soots from over-rich mixtures.

(3) Depositions on the walls of the combustion-chamber by chilled destructive combustion.

To these may be added further products of incomplete combustion brought about by the wetting of combustion-chamber walls. It also is to be considered that accumulations of soot with decomposition products of lubricating oil may be found on piston-ring lands and in back of the piston-ring grooves. In small-bore, high-speed engines many types of products can be observed deposited in the combustion-chamber and on piston surfaces when burning residual-type fuels.

Tests which have been conducted at the Caterpillar Tractor Co. for the purpose of determining the formation of aldehydes during the combustion process have failed to indicate their presence when burning distillate fuels within the speed-range limits. Soots produced by chilling influences at low jacket-water temperatures and light loads, however, are apparent but of small consequence in the lubricating-oil sump.

Fig. 14 indicates the cardinal points of mean maximum temperature in the combustion-chamber envelope. This particular diagram was taken from a cast-iron piston. When using an aluminum piston, the central piston-crown temperature drops to 550 deg. fahr.; the upper ledge of the piston to 530 deg. fahr.; temperature at the top ring land is 493 deg. fahr.

A number of experiments have been conducted with various types of piston crowns producing wide ranges of temperature of the piston surface adjacent to the combustion-chamber envelope. These temperatures of themselves have not directly influenced products deposited in the ring-belt section. The temperature of the ring belt, however, has had a decided influence on producing decomposition products of lubricating oil. This condition varies widely with lubricating oils as to their base stock and type of treatment. This point can be more readily discussed by the use of Figs. 15 and 16 showing a simple type of oxidation test in which products of decomposition of lubricating oils are produced by heat in an open atmosphere. The temperature gradient for a cast-iron bar is shown on each figure in relation to the type of products produced. In the case of Fig. 15, products of decomposition were produced within the range of 380 deg. fahr. to 620 deg. fahr. The illustration is not colored and does not show the variation from light yellow at the lower temperature through the orange and brown lacquers to the intense black carbon deposit at the higher temperature position. A lubricant of this character permitted the formation of lacquers and resins sufficient to produce aggravated ring-sticking in less than 150 hr. of operation.

On the other hand, an entirely different base stock of lubricant, treated in an entirely different manner, developed the minimum of products as exhibited in Fig. 16. Here the range of sparsely collected decomposition products were formed within a temperature range of 430 deg. fahr. to 575 deg. fahr. This figure illustrates the narrow temperature range within which such products were collected and also the reduced quantity which would naturally be expected to influence ring-sticking tendencies.

Referring again to Fig. 15, it is to be conceived that a cast-iron piston operating with a ring-belt temperature below 380 deg. fahr. or above 620 deg. fahr. would produce no difficulties from ring-sticking on this lubricant. Such a situation is beyond the realm of practical consideration for engines operating at variable load and variable speed.

Fig. 17 shows a piston removed from an engine after 1000 hr. of operation when using a lubricant of selected base compounded to minimize ring-sticking tendencies.

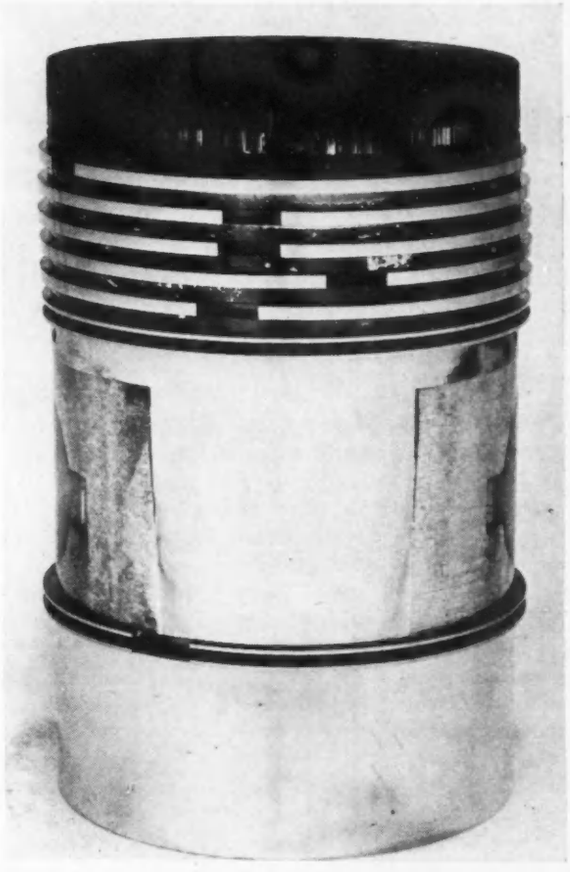

Fig. 17 – Appearance of Piston after 1000-Hr. Operation on Special Compounded Type of Diesel-Engine Lubricating Oil

It has been proved further that, within the range of overhead-distillate fuels, comparable conditions have been observed to those shown in Fig. 17, provided the same lubricant was employed. The use of compounded lubricants permits of operation over wide speed and load ranges without the necessity of thermostatically controlled ring-belt temperatures, provided the water-jacket temperatures are within the range of 150 deg. fahr. to 200 deg. fahr.

The Boerlage and Broeze paper has revealed much valuable data and is a stimulus to the further study of the chemistry of combustion as influenced by engine design and fuel properties.

The Determination of Ratings for Transport Aircraft Engines

By R. F. Gagg

Assistant Chief Engineer, Wright Aeronautical Corp.

THE objective in determining an engine rating is to establish the limiting values for the variables in operating procedure which permit a maximum of utility in power output and economy of fuel consistent with requirements for safety and durability in the class of service for which the engine in intended. The obvious safety requirement is that no interruptions to service shall occur due to engine trouble when operations are conducted in the manner established by the rating tests.

This paper consists of a discussion of testing methods to be used for the determination of engine ratings, and some suggestions for the adoption of a uniform procedure. Standardization of engine-rating procedure is thoroughly in accord with the established policy of the American aircraft industry as a means of insuring a maximum of safety to air-transport passengers.

American aircraft and engine-testing requirements are more rigorous than those of many of our foreign competitors, and it is for this reason that American products have established a world-wide reputation for safety and durability. If this leadership is to be maintained, technical standardization and advancement must be synchronized and coordinated so that each activity supplements the other.

NEARLY all high-performance aircraft engines are intended for service either in transport airplanes, or else in short-range racing or military ships in which a maximum of power output is the governing consideration. The requirements for safety and durability in the latter class vary widely with specific objectives and circumstances, and engine ratings also depend on the same variables. Study of methods for determining ratings, therefore, is directed largely toward that other and more numerous class – transport engines. Most of the aircraft now in service are powered by spark-ignition gasoline engines, and this discussion is limited to consideration of that type. The powerplant problems of the transport airplane appear to be most difficult in the case of the long-range flying boat. The engines which meet the requirements of that service are generally useful elsewhere in the transport field, and the rating procedure outlined for such a typical case is also applicable, with minor changes, to special problems such as are encountered in the racing engine.

The objective in choosing an engine rating is to establish the limiting values for the variables in operating procedure which permit a maximum of utility in power output and economy of fuel consistent with requirements for safety and durability in the class of service for which the engine is intended. The obvious safety requirement is that no interruptions to service shall occur due to engine trouble when operations are conducted in the manner established by the rating tests. Conscientious inspection and maintenance service is assumed as a self-evident necessity. With such external inspection as may be provided conveniently at intervals of about 25 hr., the engine should operate without material service work for 300 to 500 hr. between shop inspections and overhaul. With some replacement of worn parts, a total of at least 2500 hr. of service life can reasonably be expected, and there are numerous examples wherein this value has been doubled.

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 12, 1937.]

Kinds of Ratings

In considering the utility of conventional air-transport engines the power available for take-off is frequently a very important datum point, especially in the case of the flying boat. This rating determines the maximum load which can be put in flight without an assisted take-off, and the same rating may be used in a flight emergency but, in either case, a time limit customarily is specified. The duration and frequency of its use necessarily must be left to the discretion of the pilot.

For flight schedules where a maximum rate of climb is desired, or where partial failure of the total powerplant makes it necessary to increase the engine output to a maximum safe value for a protracted period, the rating termed "normal rated

power" is also useful. However, this rating is not often utilized in transport service, and its principal use occurs in accelerated tests of durability in the engine-testing laboratory.

The cruising-power output actually usable during a large fraction of its working life is the real measure of the value of an engine. Utmost care in establishing the cruising operation procedure is warranted. Power output limitations for insuring durability, optimum conditions for economical use of fuel, and determination of the effects of changes in altitude and in other variables must be determined carefully.

Limitations Inherent in Design

Approximate values for the engine ratings must be selected by the engine designer in order that the proportions of a new engine structure may be designed to withstand the inertia forces encountered at an assumed maximum rotative speed, and to provide for the desired limits of gas forces and rate of pressure rise. In beginning the task of establishing a rating for an entirely new design or for an advanced version of an established type, the experimental program must accept these design limitations as valid until experience proves otherwise. Tests must be conducted to give direct experimental proof of correct procedure.

It must be assumed that all engines will be the equal of the test unit in quality of materials and workmanship. The choice of materials with regard to fatigue properties, thermal characteristics, and the like, must tentatively be accepted as correct.

Judicious use of flexible couplings and dampers will largely eliminate detrimental resonant vibration. However, such vibration phenomena cannot always be predicted from theory or experience in the present state of the art, and they occasionally provide a shocking surprise to the mathematical augurs who predict their future characteristics by scrutiny of the entrails of the engine. The use of sensitive electrical seismographs in the determination of vibration characteristics under conditions of actual operation appears to be the best available means for study and control of vibration characteristics.

Laboratory Tests for Performance

In beginning full-scale tests of a new engine type or an extensive redesign of an existing model, the procedure is simplified greatly if previous single-cylinder testing has been utilized to provide approximately correct information on best spark timing, peak pressures, and volumetric efficiency. Separate tests also should be conducted to determine that the supercharger capacity is correct for the size of the engine, and that its thermal efficiency is acceptable. The results of such tests require careful interpretation because of unavoidable differences between single-cylinder and multicylinder engines in mechanical efficiency, fuel metering, induction-system characteristics, and other operating conditions. For this reason, most of the data in this paper are presented for multicylinder engine performance.

A reasonably complete dynamometer calibration of performance characteristics obviously is required as a basis for the rating. This calibration should be accomplished with a fuel-air ratio giving approximately maximum power output and with a non-detonating fuel in order to eliminate one possible source of uncertainty. The use of such fuel is a real economy because it permits determination of maximum power mixture strength and optimum spark timing for high power output, as well as the limits of fuel economy without auxiliary hazards. The calibration should cover the range of operating conditions likely to be encountered in service, including cruising at low power output for long range operation. Some items of performance data for a typical transport engine, the Cyclone Series G, are shown in the figures.

It is necessary to determine the relations between horsepower, engine speed, and minimum obtainable specific fuel consumption for cruising in order to obtain the best combination of propeller efficiency and low fuel consumption. These tests may be limited to the range from about 60 to 85 per cent of the engine speed used for take-off, and from about 25 to 60 per cent of the take-off power output. In this power range the ignition timing materially affects the minimum obtainable fuel consumption, as shown in Fig. 1.

It will be noted that one setting of the ignition timing provides very nearly optimum results through a wide range of operating conditions and, because of its simplicity, this arrangement is generally adopted. It is desirable to obtain the performance data for changes in ignition timing for cruising operation, and to correlate this information with high-power-output peak-pressure data before definitely choosing a setting and completing the test results.

Selection of Fuel and Compression Ratio

With an outline of engine-performance characteristics established, a review of the data is desirable to determine whether the compression ratio or other characteristics which may be varied readily should be altered to obtain the desired objectives. This review should include a study of the economics of the fuel problem before reaching a decision to proceed with endurance proof-testing to establish the selected ratings. The results of this study will vary considerably with the cruising-power and fuel-economy requirements imposed by the intended service. Unless unusual circumstances, such as necessity for extreme range with the possibility of an assisted take-off, permit a drastic sacrifice in maximum safe power at sea level, the most useful compression ratio for relatively large engines using high-octane fuels appears to be in the range from 6.5:1 to 7.5:1. The higher ratio generally will be chosen for the application where payload returns justify the use of the more expensive fuels. Fig. 2 shows, for the same engine as before, characteristic changes in minimum obtainable specific fuel consumption for variations in compression ratio with otherwise substantially constant operating conditions at moderate power output. Choice of the most useful compression ratio with due regard for its effects on numerous dependent variables, and also for the cost of the fuel, is a difficult problem which often must be decided without the assistance of

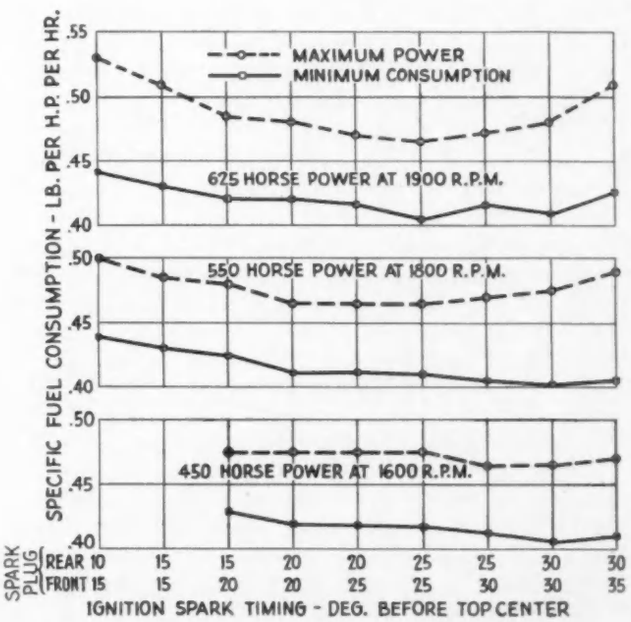
SPECIFIC FUEL CONSUMPTION - LB. PER H.P. PER HR.
.55
.50
.45
.40
MAXIMUM POWER
MINIMUM CONSUMPTION
625 HORSE POWER AT 1900 R.P.M.
550 HORSE POWER AT 1800 R.P.M.
450 HORSE POWER AT 1600 R.P.M.
SPARK PLUG
REAR 10 15 15 20 20 25 25 30 30
FRONT 15 15 20 20 25 25 30 30 35
IGNITION SPARK TIMING - DEG. BEFORE TOP CENTER

Fig. 1 – Variations in Specific Fuel Consumption for Changes in Ignition Spark Timing

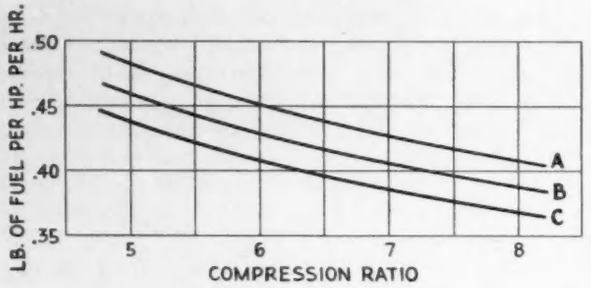

Fig. 2 – Typical Values for Fuel Consumption at Cruising Power Output

A – Practicable for Routine Use
B – Normally Attainable
C – Obtainable Under Best Conditions

complete information concerning exact service operating conditions and payload values. In such cases a compromise choice is necessary, and experience indicates the desirability of selecting the highest compression ratio which will permit non-detonating operation at the maximum power output for cruising when using that grade of fuel which includes both a high octane value and widespread availability. It will be noted that the fuel-price factor is not mentioned as a consideration because it cannot be evaluated without data on the payload value of fuel-consumption savings. The term "wide availability" implies that the existing price structure is economically practicable. Like other generalizations, this one is not exact and may be incorrect as a solution to a specific operating problem. However, the process of proving the procedure incorrect, in any case, provides the data necessary for making a correct choice for the set of circumstances being discussed, and the generalization stated earlier is a good empirical rule when incomplete data must be acted upon.

If the route length and money value of fuel savings are known, together with other pertinent data, simple equations may be written readily for the weight and cost of fuel used while cruising. An analysis of the details of a specific problem is the only clear way to a solution, and this analysis should include study of the time integral of the power required both in still air and with headwinds. If a high compression ratio is selected and abnormal conditions require cruising at increased power output, the amount of fuel quenching required to suppress detonation may nullify all of the savings otherwise obtained. Data on minimum safe fuel consumption for operation at increased power output as shown in Fig. 3 are needed in addition to similar information for cruising power output, a typical sample of which is seen in Fig. 4. This kind of data may be summarized conveniently for ready use as shown in Fig. 5.

Power Ratings

For a new engine it is clear that the maximum power output eligible for consideration as a service rating is determined by the design limitations on cylinder pressure and speed of rotation. If these limits be accepted as establishing the take-off rating, the whole operating procedure and durability-testing program may be related logically to the same values.

Use of controllable-pitch propellers and constant-speed governors makes it readily feasible to operate the engine without reference to the "propeller load curve" of earlier practice, in which the horsepower output was varied as the third power of the speed of rotation. However, it seems sensible to arrange the operating procedure (at least for the high-output range) so that the two principal kinds of stresses resulting from cylinder gas pressure and inertia forces are changed in approximately the same degree. At constant speed the power output varies directly as the mean effective pressure during the power stroke and, as shown in Fig. 6, the peak pressure in the cylinder varies almost directly with the mean effective pressure. For practical purposes, the stresses resulting from gas pressure in the cylinder may be considered as changing in direct proportion to the brake mean effective pressure. Since the principal inertia forces vary as the second power of the speed of rotation, the resulting stresses are also proportional to the square of the speed. Because the inertia forces cannot be controlled except by varying the speed of rotation, the mean effective pressures must be controlled so as to vary as

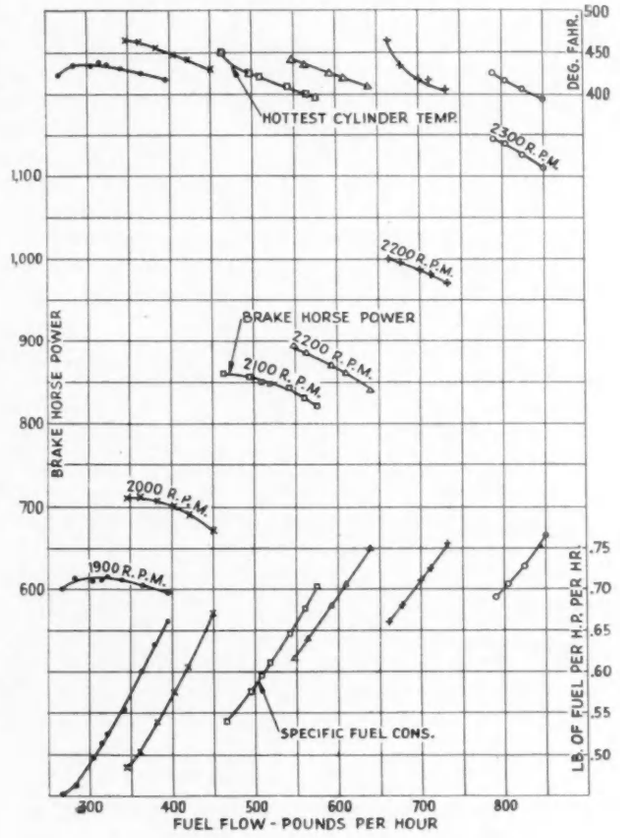

Fig 3 – Fuel Requirement Runs on Propeller Load at High Power Output (Above Cruising Range)

the second power of the rotative speed in order to obtain proportional changes in both kinds of stresses.

For any reciprocating engine:

$$\text{Hp.} \propto \text{B.M.E.P.} \times \text{R.P.M.}$$

if: $$\text{B.M.E.P.} \propto (\text{R.P.M.})^2$$

then: $$\text{Hp.} \propto (\text{R.P.M.})^2 \times \text{R.P.M.}$$

The last line defines the conventional "propeller load curve," which gives equal changes in gas load stresses and inertia stresses for variations.

For best durability at high power where material failures are most likely the engine should be "rated" for operation on a "propeller load curve" which passes through the take-off power and speed rating. Fig. 7 shows the variations in stresses for this type of operation. The "normal" power rating, which is the maximum output utilized in flight for a protracted period, generally is selected to obtain a reduction of 10 to 15 per cent in the operating stresses at the take-off rating in order to provide a substantial factor of safety for critical operating periods. The maximum power output for continuous cruising operation also should be chosen as a point on the "propeller load curve" which passes through the take-

off rating in order to obtain a balance in stress proportions. The value selected for this maximum cruising power rating is a principal factor in durability obtained in routine transport service. For a very well-proportioned engine which has an ample background of operating experience, the maximum cruising power may be about 60 per cent of the take-off power output. This value results in operating stresses approximately 70 per cent of the values at the take-off rating. If flight schedules permit, the maximum cruising power output referred to should be reduced in order to obtain a substantial improvement in durability which always follows such action, as is demonstrated amply by experience.

When long range is the principal objective, it is generally desirable to cruise at considerably less power output than 60 per cent of the take-off rating and, in such cases, the operating schedule should be arranged to obtain an optimum combination of propeller efficiency and low specific fuel consumption. Both of these objectives are served by running the engine at low rotative speeds and relatively high mean effective pressure. This procedure slightly improves the overall mechanical efficiency and results in a saving of roughly 1 per cent in specific fuel consumption for each 100 r.p.m. reduction in speed in the useful cruising range for constant power output as shown in Fig. 5. The engine durability is probably not adversely affected so long as the mean effective pressure is not greater than that established for maximum cruising output provided no detonation is tolerated, but insufficient experience is available with this type of operation to justify unqualified endorsement.

Fuel Consumption and Detonation Performance

Having selected the transport engine ratings on the basis of stress values, it becomes necessary to recheck the fuel requirements and detonation performance of the engine with the grade of fuel tentatively selected for use in flight service. Unless the performance defined in the following tabulation of specified tests is obtained without detonation, the power ratings and grade of fuel selected cannot be considered mutually satisfactory:

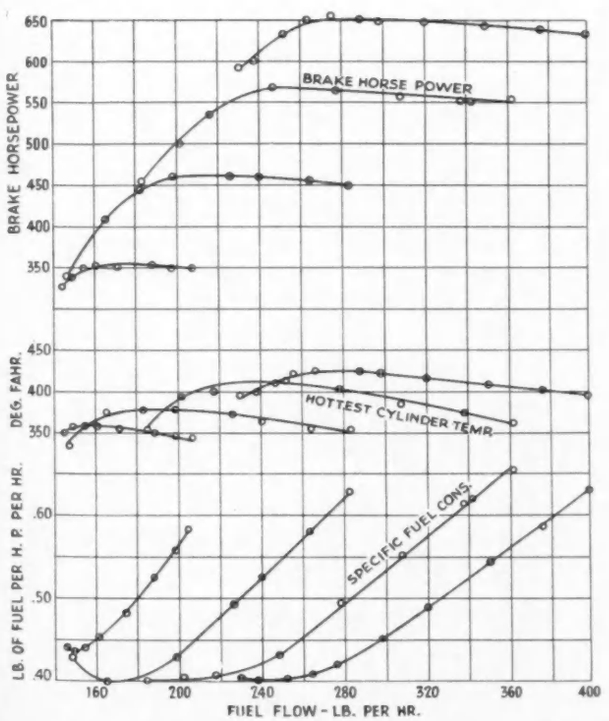

Fig. 4 – Fuel Requirement Runs at Cruising Power Output for One Crankshaft Speed (1600 R.P.M.)

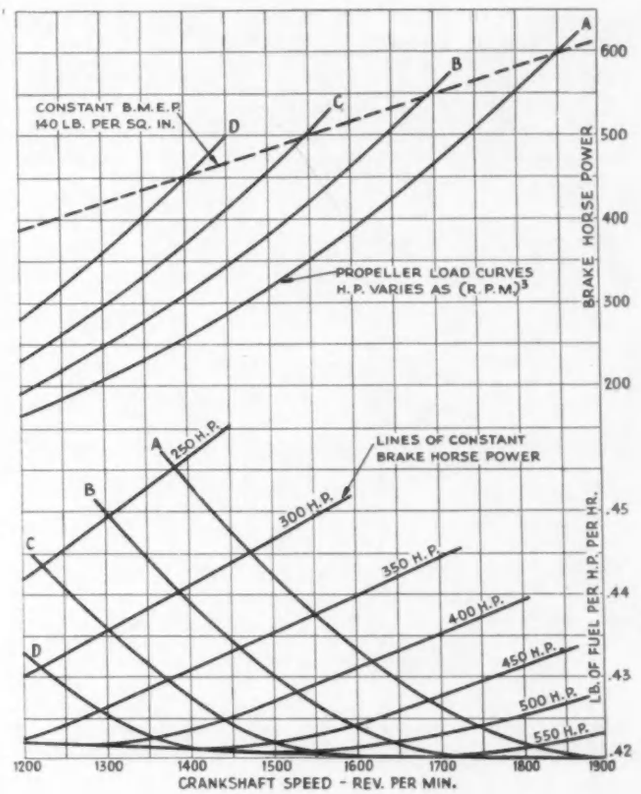

Fig. 5 – Minimum Specific Fuel Consumption Attainable at Cruising Power Output

(1) Take-off rating shall be obtained with 60 deg. fahr. carburetor-air temperature at a fuel-air ratio not greater than 0.103:1.

(2) The engine shall operate with the same speed, manifold pressure, and carburetor adjustment as specified for test (1) at 110 deg. fahr. carburetor-air temperature without detonation.

(3) The engine shall be operated at the maximum cruising power and speed with a mixture ratio approximately that for best power output, and the mixture strength shall be reduced by small increments, the throttle meanwhile being adjusted to maintain approximately constant power and speed. The mixture strength shall be reduced until the engine no longer fires regularly. The carburetor-air temperature shall be 60 deg. fahr. to 80 deg. fahr. during this test. Occurrence of detonation shall be considered unsatisfactory performance.

(4) Test (3) shall be repeated at lower speeds at intervals of approximately 100 r.p.m. to about 60 per cent of the take-off speed rating but at the same brake mean effective pressure as is obtained in test (3). If detonation is encountered, the brake mean effective pressure shall be reduced until the detonation is eliminated, and the maximum cruising output for that speed shall be reduced accordingly.

(5) The engine shall be operated at several speeds on the propeller load curve which passes through the take-off power rating at power output values as required to obtain data on minimum permissible mixture strength for endurance testing. The procedure shall be similar to that specified for test (3) described previously, except that the run shall be discontinued if detonation is encountered.

(6) For all of the foregoing tests, the differential pressure drop across the cylinder air-flow baffles shall not exceed 6 in. of water pressure. The cylinder-head temperatures shall be not less than 425 deg. fahr., or more than 500 deg. fahr. All other operating conditions shall be representative of normal practice.

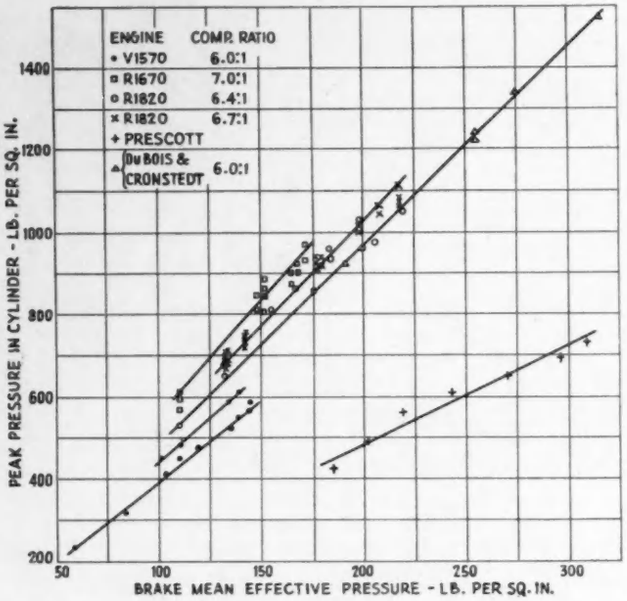

Fig. 6 – Variation in Maximum Pressure in Cylinder with Change in Power Output at Constant Speed

The test procedure outlined is intended to establish the ability of the engine to run in the high power range with the specified fuel and high air temperature with practicable mixture strengths, and to determine that the specified cruising power rating can be utilized in normal operation for level cruising with lean fuel-air mixtures without encountering destructive detonation. In selecting the test operation conditions, the suggested values were chosen to represent a reasonable and practicable compromise. It is obviously possible to choose a set of conditions which would be much less severe on the engine, or vice versa, but either extreme would tend to defeat the objective of the tests.

If detonation is encountered at the cruising-power rating, either the octane value of the fuel should be increased or the compression ratio should be reduced. The latter change is more desirable than the procedure frequently followed in the past in which the minimum mixture strength usable for cruising was limited (in theory at least) by an instruction to the pilot. Dangerous mistakes in such operation were practically unavoidable, even with the best of intentions. A reduction in compression ratio to suit the fuel being used involves, of course, some sacrifice in minimum attainable fuel consumption as is illustrated by the curves of Fig. 2. However, this sacrifice will generally be less than the fuel-consumption increase encountered in attempting to maintain a safe margin over the minimum mixture strength permitted by detonation. Any loss in maximum power output due to the lower compression ratio may be regained readily by increasing the manifold pressure, and this increase will be accomplished with a reduction in maximum cylinder pressure for the same power output.

Durability Proof Tests

In order to establish the rating thus selected for routine flight service, it is customary to run the engine for 50 hr. at normal rated power plus 10 hr. at take-off power rating (in addition to necessary preliminary running) before the engine is considered eligible for flight-testing. After this test is completed, both flight-testing under service operating conditions and laboratory testing on the ground may be carried forward concurrently.

In order to provide increased assurance of durability it is essential to continue endurance testing until a single engine has completed without failure of any part vital to operation two "type-test" schedules which each include 150 hr. of operation covering the range of operating conditions normally encountered in service, including extensive periods at high power output.

Service in transport airplanes does not normally involve more than 1 min. of operation at the take-off power rating for each hour of service, which means that a probable maximum of 50 hr. at the take-off power rating will be accumulated in a service life of 3000 hr. It is therefore desirable that a second test engine be operated for 50 hr. at the take-off rating in order to determine whether fatigue failures are likely to be encountered as a result of running in the conditions imposing maximum stresses. Since aircraft engines are not particularly useful without suitable proof-tested propellers, it is frequently convenient to combine the take-off testing with an endurance test for the propeller.

It is recommended that the details of the endurance-test procedure be regulated to include some running at each of the most difficult conditions likely to be encountered in service. For example, part of the take-off testing should be done at the brake horsepower output specified in the rating, and part at the same manifold pressure and engine speed, but with 110 deg. fahr. carburetor-air temperature. This kind of test schedule does not ignore the existence of the hazards of weather, and so on, but, instead, represents a reasonable compromise.

The test program outlined previously includes sufficient running at low speed and power to determine that operation is normal in that range, but the essential element is operation at high power output and consequently at high stresses. The 50 hr. at take-off rating involves upwards of 3,000,000 stress reversals at stress levels much exceeding those of cruising operation. If fatigue failures are not encountered in this period, it is unlikely that such fatigue failures will be en-

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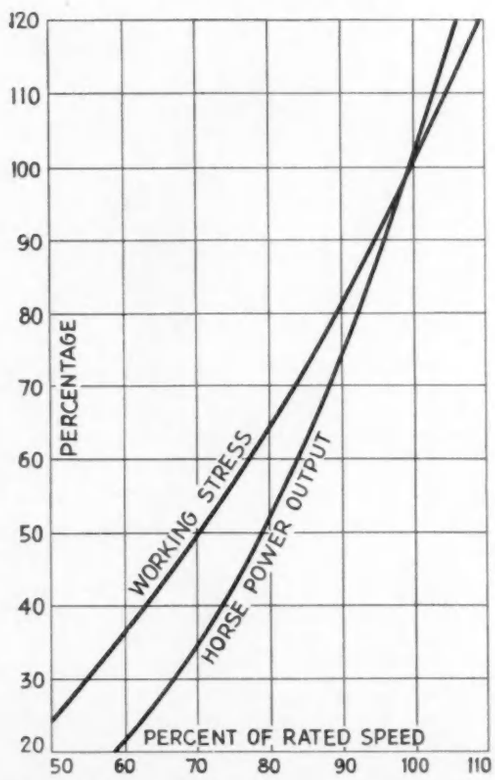


Fig. 7 – Variation in Working Stress with Propeller Load

Design Developments in European Automotive Diesel Engines

By H. R. Ricardo and J. H. Pitchford
Ricardo & Co., Ltd.

A GENERAL consideration of the particular factors which define the development of the compression-ignition engine for automotive as apart from other purposes is presented in this paper.

The various combustion systems, their respective merits, and individual characteristics in the light of the particular conditions set forth previously are reviewed.

The evolution of the general mechanical design is traced, as resulting from the requirements of the particular duty, and from experience gained under prolonged service conditions, including a survey of the most prevalent troubles and difficulties and of measures which have proved effective in overcoming them.

The paper also discusses some economic aspects of compression-ignition engine operation, and the effects of such artificial factors as fuel taxation and the distribution of suitable fuels on design and on the choice of a combustion system.

A general survey of the present uses of the high-speed oil engine for automotive purposes in Europe, including railcar developments and likely future trends, concludes the paper.

IN developing an engine for automotive purposes one is always at a disadvantage compared with those who are concerned with almost any other field of use.

This disadvantage is a fundamental one and is simply that, under automotive conditions, torque and speed need not, in practice, bear any relation to one another.

In industrial work the engine is under the control of a governor which holds it within very close limits of speed and, in marine service or even in the air, it is working over a propeller curve which means, of course, that a condition of continuous high torque at a low speed cannot arise.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 9, 1937.]

In the application of the high-speed compression-ignition engine to road and rail work during recent years, this fundamental consideration has had very far-reaching effects upon its general design, and more particularly upon the type of combustion system used. It is these effects, as they are generally to be observed in Europe, with which we propose to deal in this paper.

Naturally the term "automotive service" can cover an enormous variety of duties from small vehicle engines for use in urban districts to large locomotive engines, but we think that, at any rate, for road-vehicle work – and it is for the greater part with road-vehicle work that this paper will deal – the following conditions have come to be regarded in Europe as essential requirements:

(1) The ability to operate over a speed range closely comparable with that of an equivalent gasoline engine and, therefore, to compete closely with it on a power-weight-ratio basis.

(2) Complete freedom from exhaust smoke or smell even under conditions involving long spells of idling and light running, as in dense traffic.

(3) Freedom from combustion noise, particularly when accelerating against load.

(4) The ability to use a type of injector nozzle which is not affected in its operation by the formation of carbon, and which need not, therefore, be disturbed except at long intervals.

(5) Insensitivity to a large variety of fuels covering a wide range of ignition quality or cetane value.

(6) The ability to dispense entirely with any form of mechanically or manually operated timing device for the fuel injection pump.

Let us now examine each of these points in relation to the various duties at which automotive engines may be put.

Clearly the effective power-to-weight ratio of an engine will, to a large degree, determine its cost. That is to say that, for any given job of work, the faster an engine can be run and the higher the mean effective pressure realized, the less costly the powerplant will be. In the present state of the art, the high-speed oil engine necessarily costs rather more than its gasoline equivalent, and a large proportion of this difference is represented by the relatively high cost of the fuel-injection equipment. This equipment represents a cost per cylinder which is practically constant irrespective of the engine output and, in one instance known to the writers, the cost of the proprietary injection equipment is substantially equal to the cost of building the remainder of the engine. Admittedly, this is an extreme case, but it does serve to show how important it is, if the compression-ignition engine is to compete with its

gasoline rival, that the best possible use should be made of such costly equipment.

Weight, however, as such is more likely to depend upon artificial considerations, such as taxation based upon the weight of the vehicle. In England today weight is a very important consideration, particularly in double-deck bus work, where the arbitrary weight limits leave a scant margin for powerplant. In railcar work, weight per se is not of first importance, particularly where mechanical transmission is used, since the total weight of the powerplant is relatively small in relation to that of the rest of the vehicle as determined by buffing and other stresses. With electric transmission, however, the total powerplant weight is a considerable item, and it therefore becomes of importance to reduce the weight of both the engine and generator by running at the highest practicable operating speed.

Bulk, also, is of at least equal importance to weight in many instances and both are roughly inversely proportional to specific output.

The importance of the second requirement regarding smoke and smell is, of course, greatest in the case of vehicles operating in urban areas and in congested traffic, although it is certainly desirable that combustion should at all times be clean and complete.

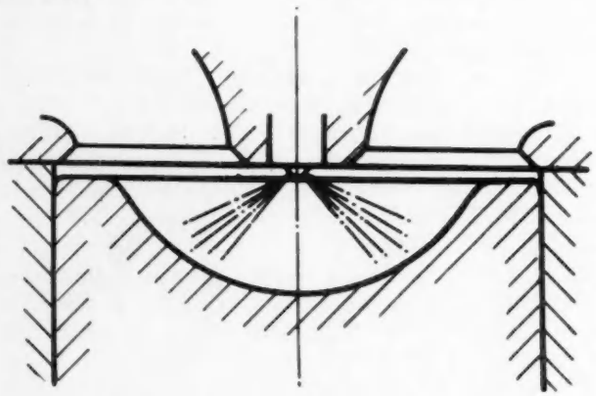


Fig. 1 – Open-Chamber or Direct-Injection Combustion System

The writers' firm has had a great deal of practical experience with this particular problem in the capacity of consultants, for the past seven years, to the Associated Equipment Co., which is almost exclusively responsible for the manufacture of London omnibuses. In such a congested city as London, with its narrow streets, the enforcement of regulations relating to smoke and smell from exhaust fumes must be, and is, rigidly carried out, and the slightest sign of smoke at the exhaust of a public-service vehicle will result in police action. There are now nearly 2000 oil-engined buses in the service of the London Passenger Transport Board, and it is certainly safe to say that, in spite of the enormously increased volume of traffic in the last few years, there has been, if anything, an improvement in the condition of the air at street level.

It is perhaps interesting to mention at this point that, after much experimental work, the separate-swirl-chamber combustion system was adopted and is now used exclusively, largely by reason of its good characteristics in this respect. Fuller reference to the various types of combustion systems now in use and their respective merits will be made later.

For heavy-goods vehicles and for railway work, the standard of exhaust cleanliness and freedom from smell, naturally, need not be so high but, nevertheless, it is improving rapidly.

The problem of reducing the level of noise and roughness in the high-speed oil engine is one whose solution has been accelerated immeasurably by the application of such engines to road and rail vehicles. The problem has been rightly tackled at its source, that is, by careful research into the combustion characteristics which can be shown to be responsible for the greater part of the audible and sensible roughness, and from the point of view of the rigidity of the engine structure as a whole.

The authors feel that, although it is a little vain to hope for the compression-ignition engine ever to compare on level terms with its gasoline counterpart for smoothness and silence, yet much can, and is, being done in this direction – quite apart from engine design – by attention to engine mounting and to the proper use of sound-absorbing materials under the hood and on other parts of the structure directly exposed to engine noise.

Taking now our fourth requirement, it may seem that to particularize on a certain type of injector nozzle for automotive duty is a rather arbitrary attitude but, in the writers' experience, it is on just such apparently small considerations as freedom from injector trouble that the success or otherwise of an engine in automotive service has depended in the past.

Apart from one or two outstanding exceptions, it is accepted among high-speed engine manufacturers in Europe that, for reliability in service, the pintle-type nozzle is second to none; the writers themselves have seen many nozzles of this type which have been in continual service with the London Passenger Transport Board and other municipal transport undertakings for more than three years – some 180,000 miles of service – without more attention than routine cleaning and without ever having been reconditioned.

The fundamental reasons for the superiority of this type of nozzle will be considered when dealing with the various forms of combustion system.

The ability of an engine to deal quietly, and in a gentlemanly manner, with a variety of fuels having a wide range of ignition characteristics, is a matter of first importance since it widens greatly the availability of supply which, in turn, must lead to a lower general selling price. A selected fuel must always cost more than that most easily available at any particular point.

Over and above these considerations looms the fact that the available supply of high-cetane fuel is steadily diminishing in proportion to the total consumption for automotive purposes. For these reasons much research has recently been directed towards this problem from the point of view of combustion-chamber design, the results of which will be dealt with in the next section.

The importance of the last point mentioned in our list of desiderata – that of the control of injection timing – is self-evident, since any step towards simplification in handling must be in the right direction.

More especially is it desirable that control of the maximum cylinder pressures should not rest in the driver's hands.

Perhaps our most useful next step would be to review the various combustion systems in use today in Europe and to see how their characteristics fit in with the requirements which we have set ourselves and which have emerged as the fruits of much bitter experience.

Open-Chamber or Direct-Injection System

In this system illustrated in Fig. 1 substantially the whole of the clearance volume is contained in some form of cavity in the piston crown, into which fuel is distributed by means of an injector having a number of small holes and usually centrally, or nearly centrally, disposed with respect to the cylinder axis.

This system has two outstanding advantages: it is certainly the most efficient type known in the sense of possible brake thermal efficiency that can be realized, on account of its ex-

ceedingly low heat losses and, by the same token, it is an extremely good cold-starter. If, however, we look at it in the light of our first practical requirement, we are at once in difficulties. The ultimate power output of any engine must depend upon its breathing capacity, and we are here deliberately forced to sacrifice valuable head area in order to accommodate our injector boss. We thus at once set an artificial limit, except in the case of a sleeve-valve engine, upon the available valve area and consequent power output. Moreover, in order to insure the intimate and rapid mixing of air and fuel which is necessary for the realization of good combustion, we must have some fairly vigorous air movement and, to do this, we must further restrict the respiratory system by masking the inlet valve and/or shaping and restricting of the inlet passage. Fig. 2 shows such an arrangement.

Again, the use of the direct-injection system in a small cylinder running at high speeds brings along another tiresome problem involving the injection equipment. A multihole nozzle must be used and, if these holes are made small enough to insure adequate atomization when idling and so reasonable stability and freedom from smell under these conditions, very high fuel-line pressures are reached at full load and full speed. As a result of this condition it becomes impossible to "unload" the fuel line down to anywhere near atmospheric pressure during the idle strokes and, as a consequence, there is fuel at a pressure of, maybe more than 1000 lb. per sq. in., "locked up" in the fuel line. Unless the injection valve can be relied upon to be 100 per cent tight – a practical impossibility over long periods – fuel will seep out during the idle strokes and will form carbon in or around the injector holes in such a way as to interfere with the distribution of the fuel both as between the various holes of each nozzle, and with the direction of the individual jets themselves. The shortcoming just described has been dealt with at some length in order to amplify the reasons for the authors' insistence on point (4) of the initial requirements set out.

These fundamental shortcomings of the direct-injection system, together with the fact that every version of the system has shown itself to be extremely sensitive to small variations in fuel ignition quality and, for the same reason, requires a large variation of injection timing with speed, either manually controlled or by the governor, in the automotive field in Europe, have shown themselves to outweigh considerably the clear advantages which the system has in fuel consumption and startability.

The Precombustion-Chamber System

Although now obsolete in England and nearly obsolete in France and Italy, this system, shown in Fig. 3, remains still in favor in Germany and Eastern Europe. It is really a development of the early Brons or Hvid engine, and differs from its original prototype only in so far as advantage is taken of the availability today of really accurate equipment for both timing and metering the fuel.

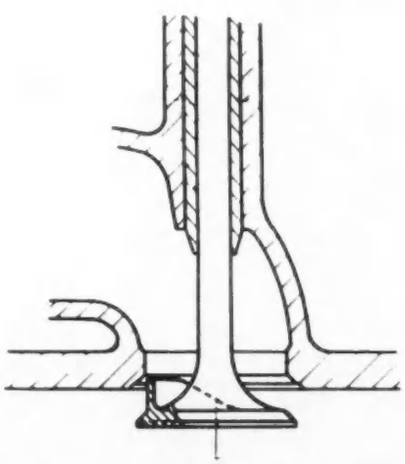


Fig. 2 – Method of Masking the Inlet Valve

In this system the fuel is injected into a cartridge or prechamber projecting into the main combustion-chamber. During the compression stroke air, compressed in the cylinder, enters the cartridge through one or more very small holes

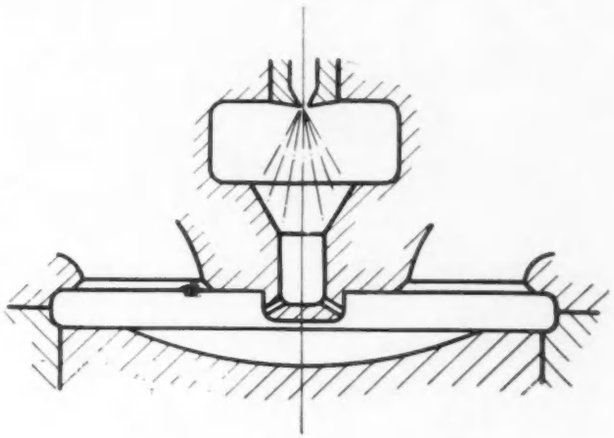


Fig. 3 – Precombustion-Chamber Combustion System

creating therein violent but disorganized turbulence. Towards the end of the compression, fuel is injected into the cartridge, the jet being aimed towards the base to meet the entering air.

Ignition of the fuel starts at or near the base of the cartridge and is accompanied, of course, by an immediate rise of pressure with the result that the burning droplets are projected violently into the main combustion-chamber, throughout which movement they are distributed in almost exactly the same manner as in an air-blast Diesel engine. No high turbulence or organized air movement is required in the main combustion-chamber since distribution is effected by the blast from the auxiliary chamber or cartridge. This system like all others has its characteristic advantages and disadvantages. In the first place, and most important of all from a practical standpoint, it can use the simple single-orifice pintle-type injector with a relatively low injection pressure. It has, further, the advantage that it is comparatively insensitive both to the ignition quality of the fuel, and to the precise timing of the injection. Also, owing to the good distribution and excellent control over the burning of the fuel in the main chamber, the maximum cylinder pressure is relatively low and is at all times under good control.

On the other side of the picture, the work done and the heat lost in transferring the air and burning fuel through small and often tortuous passages and out of the pre-chamber is very considerable and handicaps severely the power output and efficiency, both of which compare very unfavorably with either the open-chamber of the compression-swirl type. Owing to the loss of heat during transference, the engine is a non-starter from cold without auxiliary heating devices. Although the control of pressure and rate of pressure rise in the main combustion-chamber is good, that in the pre-chamber is almost non-existent, with the result that combustion knock is apt to be very pronounced. The system as a whole, has certainly had much to recommend it in the past, but today it cannot hope to compete in such countries as England, where high cost of fuel and the demand for low weight put a premium on thermal efficiency and high specific output.

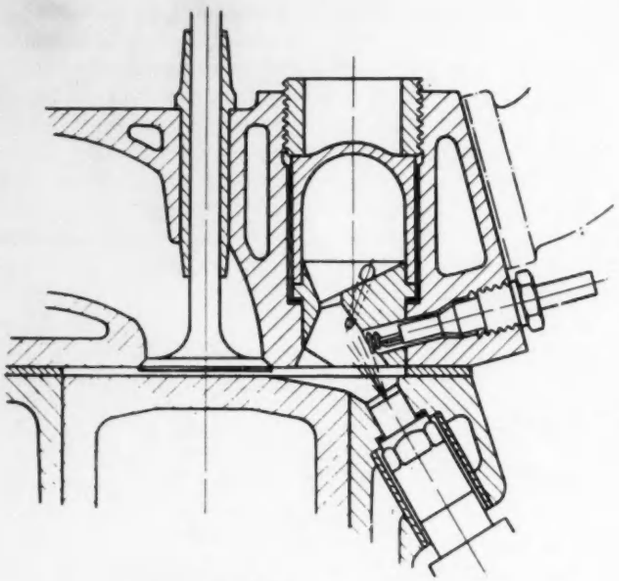


Fig. 4 – Air-Cell Combustion System

Air-Cell System

The so-called air-cell system, shown in Fig. 4, achieved some popularity in the early stages of the development of the automotive Diesel chiefly because it was claimed to be a low-pressure cycle and was adaptable easily to existing gasoline engines. In this system the combustion-chamber is divided into two compartments separated by a narrow neck, that beyond the neck being termed the air cell. Fuel is injected across one compartment, and the jet aimed at the mouth of the passage separating the two. The cycle of operations is as follows: air is compressed into both chambers, but the temperature in the first or open chamber is higher than in the air cell. Ignition takes place, or should take place, in the first chamber at or near the mouth of the air cell and, during expansion, combustion is completed by the outrush of pure air from the cell. It is obvious that, under these conditions, the maximum pressure will remain at or near the compression pressure since combustion must occur during the expansion stroke at a rate governed largely by the outflow from the air-cell, and the diagram obtained when running true to cycle would be something less than a constant-pressure diagram. It is possible to operate an air-cell engine under such conditions, but only at so low a power output and efficiency as to put it quite out of court from the point of view of practical politics. In practice, the engine appears to function on a very indeterminate cycle, and combustion takes place partly in the air-cell and partly outside. Owing to the intense turbulence prevailing in the air cell, any combustion taking place therein will be out of control and may give rise to excessive pressures and heavy combustion knock. If, on the other hand, the system operates as intended, a very severe high-temperature blast is directed straight onto the nozzle, and this characteristic has, in fact, given much trouble in service.

So far as the authors are aware, there is little to be said in favor of this system, which in Europe has now been abandoned by all except a few small manufacturers of marine or stationary engines. Its claims to favor are that it also employs a simple single-orifice injector and low fuel-injector pressure, that under certain conditions it can be made to operate very smoothly and with a low maximum pressure, and that it can be grafted very easily on to a normal gasoline engine. The disadvantages are that, if operated in a manner to give very reasonable power output or efficiency, the running conditions become very unstable; it is liable to be very noisy; there is little control over the maximum pressure or rate of pressure rise; and it is acutely sensitive both to the time of injection and to the ignition quality of the fuel.

Compression-Swirl Combustion System

This system, like that of the precombustion chamber, dates back to very early days and is shown in Fig. 5. In it the air is compressed into a chamber separated from the cylinder by one or more relatively large passages set tangentially in order to produce within the combustion-chamber a very rapid rotational air flow, the underlying principle being to set the air to find the fuel, rather than the fuel to find the air. The obvious advantage of this method is that, since the speed of rotation of the air is without practical limit and will at all times be proportional to the engine speed, the rate at which the burning fuel is supplied and replenished with fresh air is proportional to the crankshaft speed. In its practical application the combustion-chamber is spherical and the fuel is injected across it and in a direction aiming just behind the outlet of the passage. Ignition starts near the tip of the fuel spray and combustion is maintained by the continuous flow of air past it. As the temperature rises within the chamber, ignition creeps back along the spray until, towards the latter part of the process, burning is taking place close to the mouth of the injector. In order to adapt this system to the exacting conditions required today, we have found it necessary to introduce certain refinements of which the most important is the introduction of a heat-insulated member containing the transfer passage. The chief functions of this unit are:

(1) It acts as a regenerator, taking a small proportion of the heat from combustion and returning it to the air during compression.

(2) By thus raising the compression temperature, the delay period is reduced and can be controlled within any desired limits by varying the degree of the heat insulation.

(3) Since the heat capacity of this member is small and its temperature is controlled by the amount of fuel burnt, it will rise and fall with the speed and load and thus provide an automatic timing control, by reducing the delay period at high speeds, thus enabling the engine to operate at all speeds and at all loads with the fixed injection timing and at a constant maximum pressure.

(4) We have found that the characteristic smell associated with the exhaust from Diesel engines is due to the presence of heavy aldehydes; we have found also that these partial products of combustion are formed when the flame impinges against any cool surface. By aiming the fuel jet at the heat-insulated member, whose temperature at all times is fairly high, we have succeeded in eliminating almost entirely this formation – the chief source of smell.

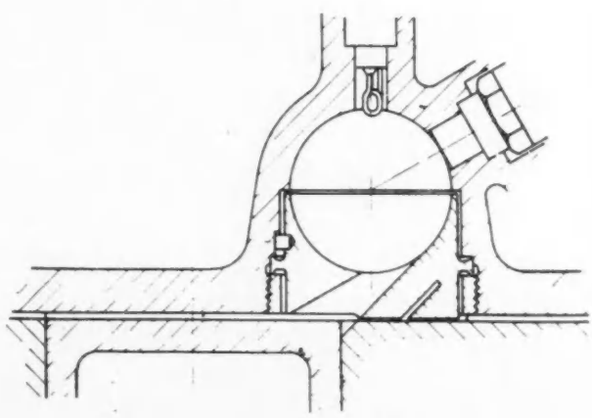


Fig. 5 – Compression-Swirl Combustion System

(5) Since the heat-insulated member is not in the path of the incoming air, it has no adverse influence on the volumetric efficiency of the engine.

This system embodying the heat-insulated passage, which for convenience we have christened the Comet type, has certain advantages. In particular neither its breathing capacity nor its combustion process is fettered, hence, in our view, it has greater possibilities in the direction of speed range and high specific output than any other system that we have yet investigated. In these respects it can compete on equal terms with the gasoline engine. It is free from smoke under all working conditions, except for a few minutes after starting from cold. As regards combustion noise, it compares favorably with the open-chamber or the precombustion types. In common with the pre-chamber type it can and does operate with a single-orifice self-cleaning, pintle-type injector and a relatively low fuel injection pressure. The intense air swirl serves also to wipe any dribble from the injector so that its performance is but little affected by leakage of the injector needle or by minor derangements in the injection pump or drawback valve.

Since the delay period can be controlled by the degree of heat insulation provided, it follows that the system is very insensitive to the ignition quality of the fuel and it will give equally good results over the range of cetane values from 40 to 60. Lastly, the hot member provides an automatic compensation of the delay period, thus enabling the engine to operate at all speeds and at all loads with a fixed time of injection.

The objections to this system are: first that, in common with the precombustion-chamber type, some work must be done and some heat lost in transferring the air into a separate chamber, even though the passage in this case is of large dimensions. To some extent this loss is compensated for by the introduction of the hot member but, of course, not entirely. For this reason, even though a greater proportion of the air can be utilized and the exhaust is much cleaner, yet the thermal efficiency falls considerably short of that of the open-chamber engine when operating at low or medium speeds. For the same reason, its ability to start from cold is also inferior. Owing to the intense heat flow at and beyond the point where the flame issues from the combustion-chamber, there is a tendency locally to overheat the cylinder-head and also to cause sludging of the lubricating oil.

With a view to overcoming these defects, we have more recently developed a new form in which the swirl chamber contains only half the total quantity of air, the other half remaining in cavities formed in the piston crown. This design has had the effect of improving the thermal efficiency, bringing it nearly up to that of the open chamber at low speeds and above it at high speeds. By removing the high-velocity flow between the piston and head in the zone near the passage, we have succeeded in reducing greatly the temperature of the head in this zone and have greatly reduced also the tendency to sludging of the lubricating oil which, from the work we have done, we feel convinced is due almost entirely to the scouring action of the flame on the oil film on the cylinder walls. A part of this oil thus contaminated is again carried below by the rings. It is therefore worth while to make every effort to avoid the tendency for the flame, during the later part of the expansion stroke, to sweep around the bore or on the top land of the piston.

When, some fifteen years ago, we started on the development of a high-speed Diesel engine, we analyzed and experimented with each of these various systems. We then came to the conclusion that the balance of advantage lay in an air-swirl system of some sort; induction swirl had the advantage of a higher efficiency, but it was difficult, we found, to eliminate smell or to make it insensitive to injection timing or to the ignition quality of the fuel. Also, except in the case of sleeve-valve or two-cycle engines, we could not obtain sufficient swirl without penalizing seriously the volumetric efficiency. We therefore resorted to compression swirl for poppet-valve engines since it appeared from the point of view both of breathing capacity and of rapidity of combustion that this system alone held out any hope of achieving the really high specific outputs and speeds which would be called for in the not-far-distant future. Typical indicator diagrams for four combustion systems are shown in Fig. 6.

It may be of interest to mention that the authors' firm has also devised another variation of the compression-swirl system in which the aim has been to achieve the best possible standard of cold-starting together with smooth running, without any sacrifice in efficiency and power output.

Both of these requirements have been achieved successfully after a great deal of work and as an indication of the standard of startability realized, it is possible to hand-start an engine of 4-in. bore and 4½-in. stroke at a temperature of 10 deg. fahr. on 45-cetane fuel and, alternatively, to start the same engine by hand at around 32 deg. fahr. on a fuel of 28-cetane value.

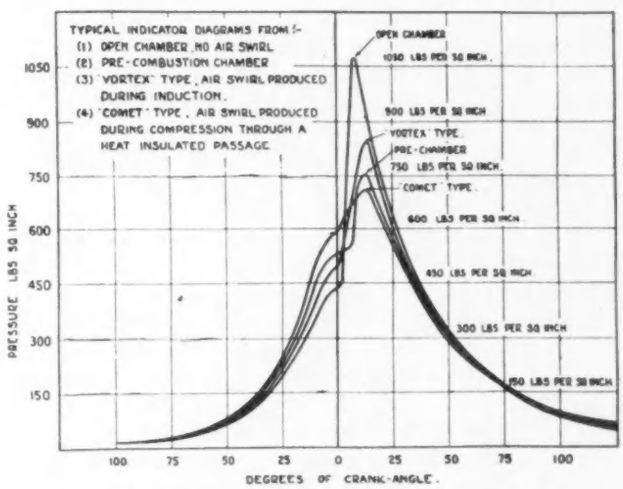


Fig. 6 – Typical Indicator Diagrams for Four Combustion Systems

This system has proved most successful in service and is now in full production by, among others, one of the largest producers of oil-engined vehicles in Europe.

A general review of the mechanical construction of high-speed automotive oil engines in Europe today and some notes on the most persistent troubles and how they have been alleviated and, in some cases, overcome, may be of interest since they represent the authors' direct experience as consultants to the majority of the leading European oil-engine manufacturers. Such a review, however, must be brief and necessarily incomplete in such a paper as this which I feel already is becoming more than a little top-heavy.

The one-piece construction of cylinder-block and crankcase as one iron casting recently has gained much ground and is clearly eventually going to oust the rival fashion of using a separate light-alloy crankcase and iron cylinder-block. The reason for this change is two-fold: first, light-alloy crankcases lately have come under something of a cloud as being answerable, through distortion both temporary and permanent, for a good deal of the main bearing trouble that has been experienced in the past by a number of manufacturers, and second, given competent design, it is possible to realize the same weight in the one-piece iron construction as when use is made of a light-alloy crank-chamber.

For road-vehicle work today a compression-ignition engine

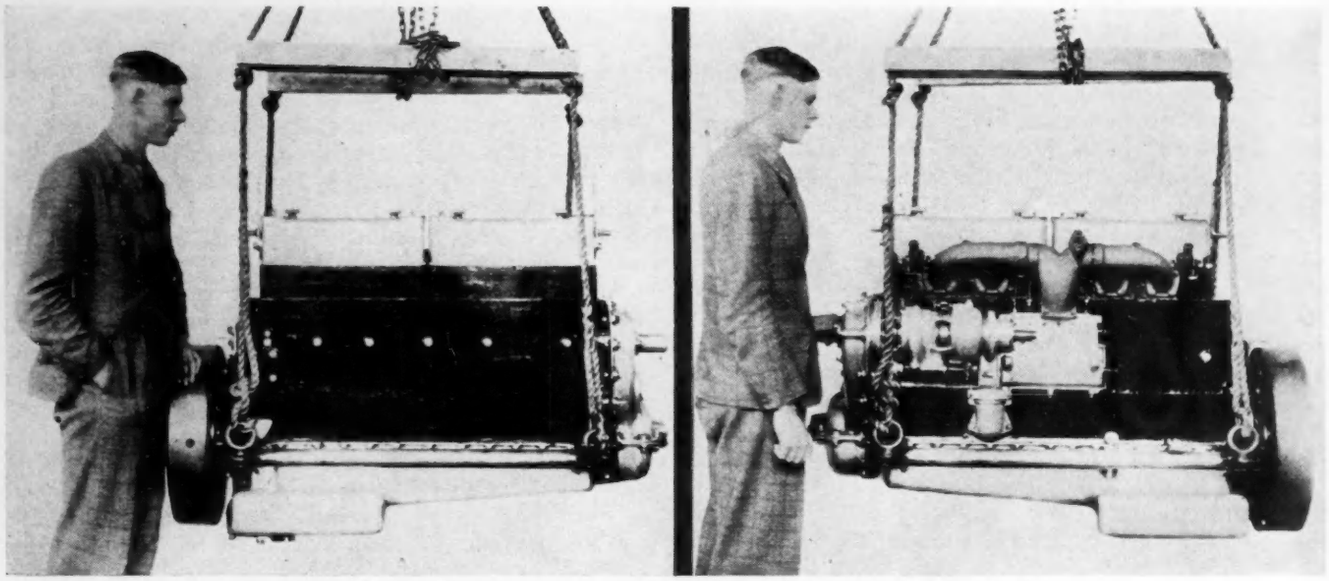


Fig. 7 – Modern Bus Engine Having One-Piece Cast-Iron Construction

should not weigh substantially more than 10 lb. per b.hp., based on the maximum power at governed speed, whether it be of 100 or 600 cu. in. swept volume. This weight to include the flywheel, but not the generator or starter motor. Fig. 7 shows a typical example of a modern bus engine having one-piece cast-iron construction.

The writers' firm has been responsible for several designs which have come out in iron construction throughout, appreciably below these weights, and have proved entirely satisfactory. With a really rigid crankcase construction and some provision for keeping the lubricating-oil temperature down to the order of 200 deg. fahr., main-bearing troubles have practically disappeared and, in several instances, white-metal bearing rings have come back into use, after lead-bronze had seemed the inevitable but unwelcome solution. Three main bearings have shown themselves to be perfectly adequate in the case of four-cylinder engines of less than about 150 cu. in. capacity, although there is certainly some sacrifice in smoothness unless the end diaphragms of the crankcase are made extremely stiff.

In the case of six-cylinder motors, the seven-bearing arrangement carries the day easily. It allows for a wide center bearing and so makes it possible to dispense with a fully counterweighted shaft which is so undesirable from the point of view of the problem of crankshaft torsional vibration to which further reference will be made later.

The value of making the crankcase structure as wide as is practicable at crankshaft level has become more and more appreciated by designers as easily the most effective step in resisting any tendency of a six-cylinder motor with an unbalanced or partially balanced shaft to "wriggle" longitudinally.

There lately has been a very marked swing towards the dry-liner type of construction as presenting less difficulty in insuring absence of distortion under operating conditions and in being free of awkward gas and water joints – frequent sources of trouble in our experience. The dry-liner engine may be slightly more expensive to build and very slightly heavier than its wet-liner counterpart, but it is in most ways a more satisfactory construction.

As regards cylinder-liner wear, this has proved itself to be a function far more of the type of service to which an engine is put than of the class of material used for the liner. In other words, except under special circumstances where the engine has definitely to work in an abrasive atmosphere and where really adequate air filtration cannot be relied upon, the life of the cylinder bores will depend primarily upon what proportion of the engine running life is spent at loads where the inner surface of the liner is above the dew point of the acid products of combustion, and only secondarily upon the character of the liner material although, of course, the greater the resistance of a material to corrosion, the better chance it will have. This point is brought out graphically in Fig. 8, for the use of which the authors are indebted to C. G. Williams, Director of Research, Institution of Automobile Engineers.

The critical temperature above which the liner ought to operate has been shown to be about 160 deg. fahr. Much is now being done by the use of thermostats.

An average wear figure can mean but very little but, for what it is worth, 0.004 in. per 1000 hr. measured at top ring level would probably be a fair figure today for European engines.

Turning now to pistons: some form of aluminum alloy has been adopted almost universally for material, used in some instances where excessive ring-groove wear has been experienced, with a cast-in iron ring-carrying belt embracing the top one or two pressure rings.

Pistons, as such, have shown themselves to be remarkably trouble-free, but it has been quite another story with piston-rings which, in many quarters, have proved serious obstacles to development, and at least one very large engine producer on your side of the Atlantic, in conjunction with some of the leading oil companies, has carried out much painstaking and costly research on the problem of ring-sticking. It is interesting to observe that in Europe, although it has shown itself from time to time as a temporary difficulty, ring-sticking has never proved to be one of the major problems in automotive Diesel-engine development. Let us briefly review the factors which, in the authors' opinion, are mainly responsible for this condition, and see what steps have been successful in minimizing their effect.

Dirty Combustion

Under this head is included not only poor combustion as indicated by a smoky exhaust at or near the full-load end of the operating range, but also the even more detrimental condition as far as ring-sticking is concerned, of a tendency

to misfire or fire incompletely when running at very light loads and high speeds.

Fortunately, as a result of recent combustion-system developments, there is now not the smallest excuse for any high-speed oil engine to operate with anything but immaculate combustion either at no load and full speed or when closely competing with its gasoline rival for specific power output, irrespective of size.

High Piston and Ring Temperature

This is a very important factor indeed, and its amelioration may be considered from two points of view:

First, if a ring is to operate at its lowest temperature under any one set of conditions, it is absolutely essential that it should seal as nearly perfectly as is humanly possible, both on its circumferential face and on its sides, and that this high standard of sealing should be realized from the first time the engine ever fires. The writers have, on several occasions, come across instances where the ring has been so badly distorted in fitting it to the piston that it has been foredoomed before ever it has *been erected in the engine*. In this respect the roundness of the liner is equally important, and there is much to be said in favor of the growing practice of finish-machining the cylinder bores after the cylinder-head studs have been fitted and tensioned against a dummy head. Any increase in the number and reduction in the size of the head studs used, of course, is beneficial in this direction. These measures also will have the effect of reducing the mean lubricating oil temperature – always a step in the right direction.

Second, given the best possible standard of oil tightness, it has been shown to be well worth while to do everything to insure the freest possible heat flow from rings to bore, and from bore to cooling water.

For instance, even at the cost of a slight increase in engine height, it is most desirable that the ring pack should not reach a point below the level of the water jackets at bottom dead-center since, owing to the angularity effect of the connecting-rod, the rings are virtually stationary at this point for an appreciable time, and so have a real chance to evacuate some heat. This measure also will have the effect of reducing the mean lubricating oil temperature – always a step in the right direction.

Lubricating-Oil Consumption

There is a great deal of evidence to show that there exists what might be termed a "pessimum" consumption above or below which ring-sticking does not occur except where the other factors are quite unusually influential.

As to the effective control of lubricating-oil consumption, practically all of what already has been said about rings and cylinder bores applies with equal force. Much has been achieved recently by working the slotted-type of oil-control ring with the absolute minimum of up-and-down clearance in the groove.

It is, of course, possible, though difficult to reduce the amount of oil passing the piston so as to jeopardize adequate lubrication but, providing that the other factors mentioned are kept under careful control during the design and manufacture of the engine, there remains a very wide tolerance between this state and the "pessimum" consumption for any particular engine.

It is interesting to see that a number of automotive-oil-engine manufacturers have taken a leaf out of the aero-engine makers' book in using taper-section rings in the top two grooves in order to overcome ring-sticking. This practice has long been applied successfully to high-output air-cooled gasoline engines where piston temperatures are of a very high order, and it has shown itself equally successful in the case of the oil engine where it, in the writers' experience, has effected a complete cure in every instance where it has been applied properly and always is recommended by them without reservation.

Perhaps the other great problem in the development of the type of engine that we are considering has been the failure of both main and big-end bearings from "shattering," which occurs almost invariably in the lower half of the former and the upper half of the latter.

Much careful thinking and a mass of experimental results have so far failed to relate these failures to any specific weakness in design or particular conditions of operation, and all that the writers feel able to do with sincerity is to set forth a few points in design and manufacture which have proved essential to success, together with one or two suggestions which may perhaps be new to some.

Bearing shells must be fitted both in the connecting-rod and in the main-bearing housings so as to reduce deformation under load to an absolute minimum. The total thickness of a bearing shell can with advantage be kept down to the minimum since clearly the further the bearing surface is from the neutral axis of the shell the greater will be the stress range imposed upon it by any deformation. Much damage can be done in pulling up bearing bolts, and these bolts should not be of such a size in relation to their bosses as to make deformation of the housings and shells possible, and hexagons of a size smaller than standard for the bolt size can be used to advantage.

The mean temperature of a white-metal bearing is undoubtedly the most influential single factor in determining its ultimate life.

The resistance to fatigue of babbitt metal is reduced enormously by allowing only a relatively small rise in temperature, and anything expended in improving this condition will be amply repaid.

Providing always that the contact between the back of the bearing shell and its housing is better than that between shell and crankshaft – and no lower standard than this can be contemplated – the main bearings are relatively well situated since they have the whole structure of the engine as a heat reservoir and radiator. The big-ends, on the other hand, depend entirely on the oil in circulation to take away their heat since clearly no flow of heat *up* the connecting rod of a high-speed engine can be imagined. It is necessary, therefore, to realize the highest possible flow of oil through the bearings consistent with a reasonable oil consumption, and to maintain the temperature of this oil at the lowest practicable figure.

One solution of this particular problem which has in the authors' experience proved entirely satisfactory is shown on Fig. 9.

In this arrangement a large lubricating oil pump is used

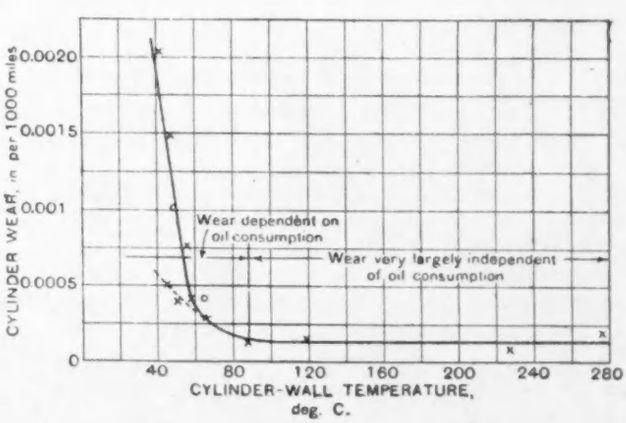


Fig. 8 – Cylinder Wear in Relation to Wall Temperatures

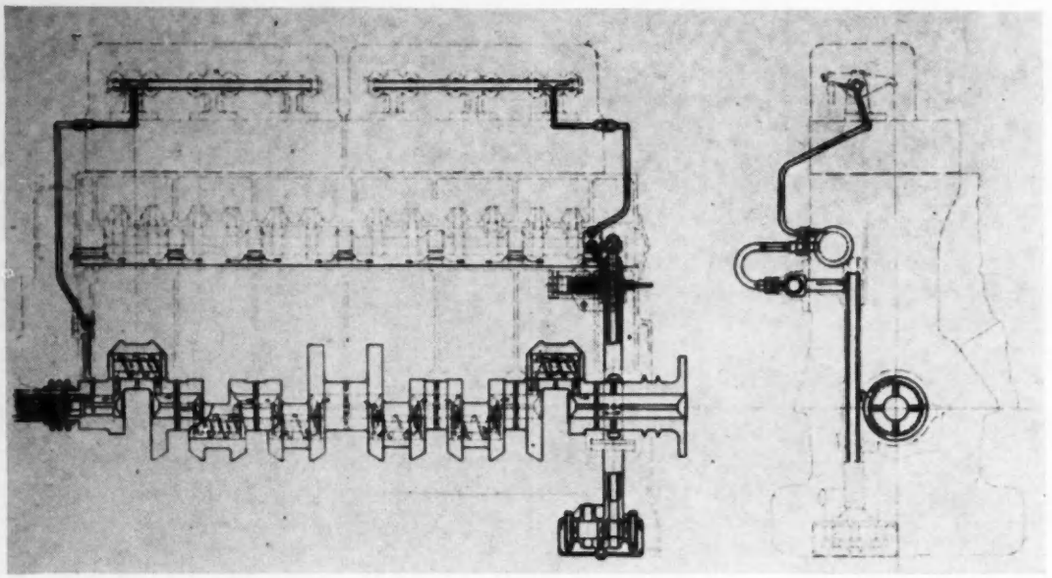


Fig. 9 – Lubricating Arrangement of Six-Cylinder Compression-Ignition Engine – 4¾ In. Bore, 5½ In. Stroke

and the whole output from this pump is circulated through the full length of the hollow crankshaft. A relatively small amount of oil is allowed to escape into each main and big-end bearing to support lubrication, whereas the heat dissipation is dealt with entirely by the large volume of oil flowing through the shaft.

A relief valve at the forward end allows the oil to escape onto the chain and so to drain back to the sump. It will be noticed that light aluminum spirals are fitted in the crankpins in order to keep up the velocity of the oil against the inner surface of the pins.

In the type of engine that we are considering, having a conventional lubrication system, not less than 1½ gal. of oil per b. hp-hr. should be circulating at full load, and it always should be possible to keep the oil temperature down to around 180 deg. fahr. This control will generally insure that the bearing temperature will not far exceed, say 220 deg. fahr.

The use of lead-bronze and cadmium-silver as bearing material has of course made possible much higher operating temperatures, but at the same time, it has made necessary a considerable increase in bearing clearance of the order of 40 per cent, with its consequent aggravation of the oil-consumption problem. The use of these materials also has forced upon manufacturers the expensive and sometimes treacherous hardened crankshaft.

The writers recently have been much impressed by the exemplary behavior of "Satco" bearing metal in certain experimental engines in which specific bearing loadings and rubbing velocities are quite unusually high, and in which almost every other type of bearing has failed in a relatively short time. One of these units having a bore and stroke of 6 in. by 6.6 in. runs at 2500 r.p.m. mostly at a b.m.e.p. of between 160 and 170 lb. per sq. in., giving a big-end loading of the order of 3,500 lb. per sq. in. With a bearing of so high a lead content an unhardened shaft is, of course, entirely satisfactory.

The importance of the avoidance of any high local stress concentrations in bearings is often overlooked, and it is always surprising to see engines in which, for instance, the crankshaft oil-ways are drilled parallel to the respective webs so that the surfaces of the pins and journals are broken by these holes just at the points where the full pressure loading is most directly applied. Oblique drilling of the webs costs no more, and is a complete answer to this criticism.

Another instance – the breaking of the upper surface of the big-end bearing right in its most heavily loaded zone in order to provide pressure lubrication to the small end. Such provision has, in high-speed engines, proved to be entirely unnecessary. If, however, any qualms are still felt by the designer about this point when starting from cold, the scheme shown in Fig. 10 can be recommended with complete confidence and is now in wide use.

Here the drilling in top of the small end of the connecting-rod is extended downward into the shank of the rod itself, thus forming a small oil well which is filled during the initial erection of the engine and will be found to remain full indefinitely by replenishment from the oil collected in the beveled mouth of the hole.

This oil can reach the well by passing around a shallow groove in the back of the rod bushing.

Another rather interesting instance of what can be done by the relief of high stress concentrations is perhaps worth mentioning. Arguing that complete rigidity of the arch of the rod is unattainable, one of our largest licensees has recently more than doubled the minimum mileage before failure of the top half of his white-metal bearings by drilling a 12-mm. hole (on a 48-mm. big-end) clean through the most robust part of the rod parallel to the crank pin, immediately above the arch of the big-end. This measure, in effect, transforms the top half of the bearing more nearly into a strap, and so avoids the high concentration of loading in the center, where trouble almost invariably starts. In their efforts to combine the load-carrying capacity of lead-bronze with the dirt-accommodating properties of babbitt, a number of manufacturers are using successfully a composite bearing with lead-bronze in the top half and white metal in the cap of the big-end and, of course, vice versa in the main bearings.

This arrangement, given a reasonable degree of oil cleanliness, allows the use of an unhardened crank.

Before ending this section on mechanical design we must deal, necessarily incompletely, with the very important subject of crankshaft design. Today the scantlings of any high-speed automotive engine crankshaft are, or certainly should be, determined primarily by the consideration of keeping the major critical speed of torsional vibration – the 6th order in the case of a six-cylinder motor – outside the running range of the engine as determined by the governor.

Let us just review, as briefly as possible, what steps can be taken in the design of a shaft to insure that its natural frequency will be as high as possible.

(1) The shaft must be as short and as stiff as is consistent with adequate bearing area. This requirement means that cylinder centers must be closed up as tightly as we know how. How close we can get these centers almost invariably will turn on the valve diameter necessary for the power output required.

(2) We must have the metal in the shaft so disposed as to give the greatest stiffness consistent with the lowest moment of inertia value. This requirement means oval webs as the

often-too-costly ideal which, in the case of a seven-bearing shaft, can be relatively narrow in side view, but as wide as possible axially.

(3) The weight of the piston and connecting-rod assembly must be kept down to a minimum consistent with strength. Piston weight is today usually always at or near the practical minimum and the fact that, for automotive engines, it must pass through the bore does not allow of any wasteful distribution of metal in the big-end.

(4) Counter-weighting must be reduced to the absolute minimum consistent with the bearing area available and the general stiffness of the engine structure since its effect on the natural frequency of the shaft is clearly wholly bad.

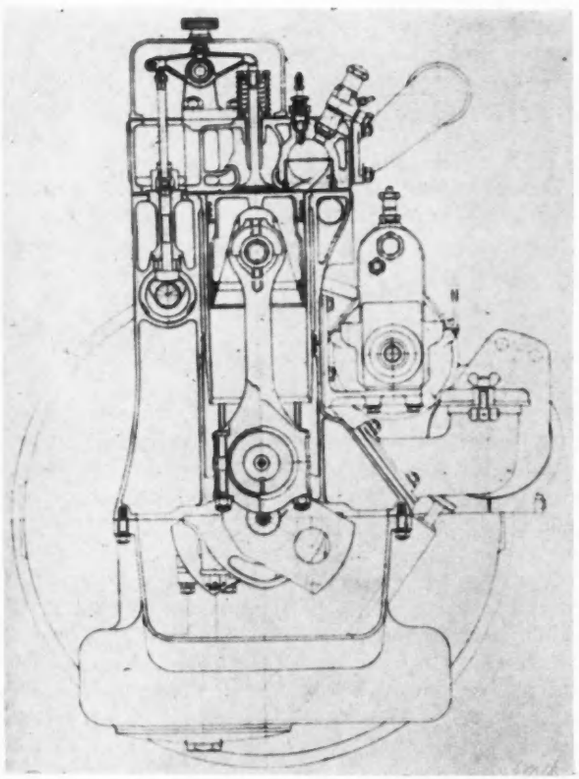


Fig. 10 (*above*) – Cross-Section of Six-Cylinder Compression-Ignition Engine – 4¾ In. Bore, 5½ In. Stroke

Where it is felt that an entirely unbalanced shaft cannot be tolerated, "half-balancing" has proved quite a satisfactory compromise.

By "half-balancing" is meant the fitting of weights to the inner webs of the two center throws in a six, and to the inner webs of Nos. 1 and 6 lines at 120 deg. to the webs. See Fig. 11.

Even if we do all possible along these lines, it is usually extremely difficult to push the shaft frequency of the 500 cu. in. size of engine above say 12,000 r.p.m. which gives us our 6th order at 2000 r.p.m. or uncomfortably near the useful maximum speed of such an engine. As is always the case, however, when we come down in engine size to say about 300 cu. in., the picture brightens immediately and there is no difficulty in getting the shaft frequency up to around the 18,000 r.p.m. mark and consequently our 6th order up to about 3,000 r.p.m., or well above the speed at which we should, for other reasons, wish to govern such an engine.

In larger sizes of engines the 4½ order also comes into the picture, but much of its sting can be removed by a re-arrangement of the firing order.

Torsional dampers, of course, can be useful so long as they do not seize unbeknownst to the operator, and so make conditions infinitely worse than before.

Before ending this section on mechanical problems it should be mentioned that very great strides recently have been made in controlling the temperature of the cylinder-head face where it is exposed to combustion. This advance has been achieved in part by directional circulation of the cooling water at high velocity, and in part by careful attention to the thickness of the head deck at the important points.

The vigorous circulation of water directed by pipes or cored passages in the head to the high-temperature zones prevents the formation of hot pockets and of steam bubbles, and so minimizes the deposition of scale. A circulation at full speed of not less than 15 gal. per b. hp-hr. should be realized. Not only have these measures quite eliminated head-cracking; they also have put an end to many obstinate cases of valve-seat distortion. A small point, but an important one, is that all injector holes must be bored on both diameters *and* faced for the joint at one operation.

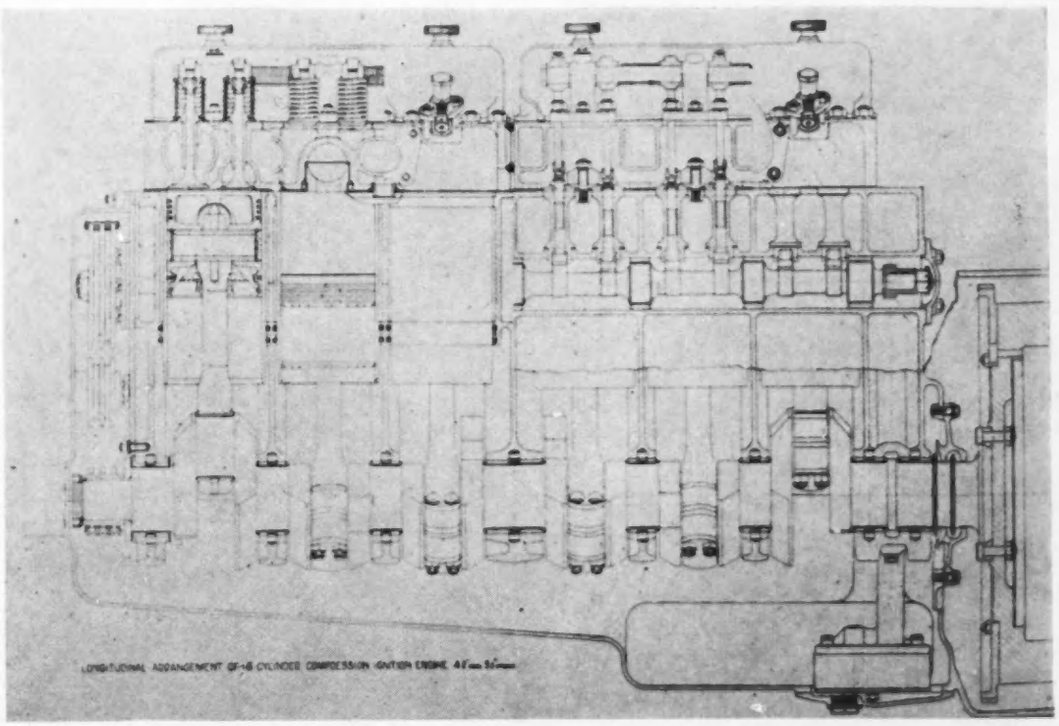


Fig. 11 (*right*) – Side View of Six-Cylinder Compression-Ignition Engine – 4¾ In. Bore, 5½ In. Stroke

This section may seem only to have touched inadequately on many vital points, and to have omitted others entirely; such treatment is unavoidable in a review like this.

The authors intended at this juncture to consider some of the economic aspects of automotive oil-engine development but, as there remains much to be said of present and likely future technical developments in Europe and being engineers and not economists, they feel compelled – and more than a little relieved – to pass over this phase as briefly as possible, or rather to combine it with the last section of the paper dealing with the general position in Europe today and likely future trends.

The growth of the automotive Diesel engine in Europe has been extraordinarily rapid, most of all perhaps in England, for we were slow to be convinced and started late but, during the last few years, it has made extraordinarily rapid strides in our country. So much so, in fact, that the sudden and unexpected imposition of a heavy tax on Diesel fuel (which has brought the price to the same figure as gasoline) has not, to our surprise, checked appreciably its upward progress. No corresponding tax exists in other European countries where the price differential remains largely in favor of Diesel fuel; yet, despite this handicap, the rate of change from gasoline to Diesel engines is more rapid in England than in any other country. The immediate effect of the heavy tax has been, of course, to put a high premium on fuel economy and, for a short time, it appeared as though there might be a general swing-over in favor of the open-chamber engine. Nearly every operating company experimented with open-chamber engines but, during the last four months, most have returned again to the indirect type on account of its lighter weight, higher speed average, and lower maintenance costs, and – last but by no means least – its absence of smell which is so important in city service.

It is difficult to obtain accurate figures for the total number of Diesel vehicles operating in Europe, or even in England, and published figures, said to be authentic, differ so widely that we do not feel justified in quoting them. We do know, however, and for certain, the output of "Comet"-type engines in service since this figure is supplied in the royalty returns from our various licensees and certainly will not be overstated. Up to the end of December, 1936, the total horsepower of Comet engines in service had reached 2,000,000, and it is still rising rapidly.

Although there was a swing of opinion in favor of the open-type chamber after the announcement of the tax, it was one of opinion rather than of purchase. The imposition of the tax in our country had also another effect in that it discouraged the practice of changing engines from gasoline to Diesel in existing vehicles – a very good thing in any case – and it put a check also, though probably only a temporary check, on the production of small engines for the lighter classes of vehicle.

It is difficult also to obtain accurate figures for running or maintenance costs since such figures are apt to be colored by the enthusiasm or otherwise of the operator, but the London Transport Board which today operates a fleet of nearly 2000 Comet Diesels and also about 3000 gasoline buses has recently stated publicly that the maintenance costs of the two are now about exactly the same, while the mileage per gallon of the Diesels is a full 100 per cent greater, despite a very rigid routine tuning of the gasoline engines. The Board has stated further that the whole of the London Omnibus Service will be converted to Diesel fuel in as short a time as possible.

We have seen that in England the imposition of a truly staggering tax on fuel oil for automotive uses caused but the smallest check in output. That is to say that, for the great majority of uses, the oil engine can justify its higher initial cost on the 2 to 1 advantage of fuel consumption alone. How much stronger then is the position of the small compression-ignition engine in, say, Italy and France where there exists still a price differential on the respective fuels of the order of a 2 to 1 advantage in favor of the Diesel fuel, making an overall advantage of 4 to 1 on a fuel-cost-per-mile basis.

In actual fact, the confidence of these two countries in the future of the automotive Diesel is enormous, and is clearly shown by the steady call for smaller and smaller engines running at yet higher speeds.

As an illustration of this point one of our French licensees who, incidentally, is one of the largest individual producers of vehicles in Europe, is now in steady production with a four-cylinder engine of 107 cu. in. capacity which develops 45 hp. at 3800 r.p.m.; and which already is in extensive use in delivery vans, taxicabs, and commercial travelers' cars. In urban delivery service and taxicab work, where a lot of light running and idling is involved, these little engines are showing a fuel consumption of less than one-third of that of the equivalent gasoline motor, or a fuel cost of around one-seventh of the latter.

Clearly, under these circumstances and where the whole level of fuel costs is relatively high, as in France today, the extra cost of the oil engine vehicle is justified easily, as is shown by the fact that this particular firm cannot yet begin to keep pace with the orders received for these small oil-engined vehicles.

As far as railway work is concerned, we in England lack practical experience, since the political importance of the use of coal, with the exception of a few very small shunting locomotives and some rather interesting railcars, has ruled out the oil engine.

We can see, however, clear pointers to the future among our Continental licensees and, in Italy in particular and in France to a lesser extent, oil-engined rail coaches are being made on mass-production lines.

It is interesting to note that, of the three principal builders of this type of railcoach with whom we are connected, two have arrived at a very similar solution, and one which the authors have long thought to be logical.

For powerplant, two perfectly standard truck engines are used, giving a maximum output of between 130 and 150 hp. each. In one case these engines drive through a Sinclair hydraulic coupling to a Wilson epicyclic type of gear box, and thence each to its respective bogie.

In the other instance, a perfectly conventional truck transmission system is used between each engine and its bogie, the two clutches and gear-change mechanisms being operated and synchronized pneumatically.

The whole attraction of these two systems is that in each case, except for minor detail modifications, all the components involved are standard production articles as made in large numbers for trucks and buses. The cost of these vehicles comes out, therefore, at only a fraction of that of the equivalent coach having specially designed powerplant and transmission. Of the problem of the large Diesel locomotive, I propose to remain discreetly silent since it is in your country that real progress here has been made in recent years, and I am over here anxious to learn all that I can in this very specialized field.

In conclusion, the authors feel confident that the further evolution of the high-speed automotive oil engine will not be unlike that of living things – the light, compact structure made possible by stamina at a high level will, except under very special circumstances of environment, always oust the heavy and the cumbrous, and the development of the omnivorous quality will always show a sweeping advantage.

Value of Octane Numbers in Flying

By D. P. Barnard
Standard Oil Co. (Ind.)

THIS paper represents an attempt to illustrate the values of octane-number improvements in aviation gasolines in terms of increased earning power of current-type transport airplanes when proper provisions have been made in the original designs. The procedure consists in computing the change in earning power of a gallon of gasoline when octane-number changes are reflected in altered fuel consumptions or take-off load capacities. In general it appears that:

(1) Depending upon operating conditions the potential revenue earning power of one gallon of gasoline may be increased from 2 to 8 cents per octane-number improvement.

(2) If the octane-number improvement involves a decrease in energy content, the apparent improvement must be discounted by about 2 octane numbers for each reduction of 1 per cent in heat content below that of gasoline.

(3) The economic necessity for high-octane number fuels is particularly apparent when long-range operations are involved.

(4) It is evident that the earning power of octane-number improvements is so great that, within practical limits, cost cannot influence the trend toward higher octane numbers to any appreciable extent.

THE following study has been made as an attempt to evaluate changes in the knocking characteristics of aviation fuels from the point of view of their effects upon the earning power of engines in transport airplanes of current type. It is recognized that, for some years past, there has been a trend upward in the octane-number requirements of aviation fuels and that this trend has been accompanied by the use of transport airplanes of increased load-carrying and speed characteristics which, in turn, have been made possible by the availability of engines of improved performance. Further, it is generally appreciated[1, 2] that impending airplane designs will call for still higher octane number fuels in order to permit the use of powerplants capable of carrying still larger loads and attaining higher cruising speeds than are possible at the present time.

However, it is apparently not always clearly understood, especially by those not directly connected with the aviation industry, just why higher octane number fuels are so highly essential. In many instances the only apparent result has been increased cruising speeds which, in turn, have been reflected by time savings of, say, the order of 15 or 20 min. for the distance between Chicago and New York. Such gains do not appear to justify any large increases in fuel costs. It is an actual fact, however, that engines taking *full* advantage of octane-number increases make possible very real increases in the earning power of the equipment. It is the purpose, therefore, of this study to indicate as reliably as possible the earning values which can be achieved by octane-number improvements.

Method of Calculating Revenues

Heavier-than-air type aircraft depend upon the availability of sufficient power to produce the accelerations necessary to attain flying speeds within the confines of airports of practical dimensions. This consideration has resulted in the land-transport type airplane being held to fairly definite limits of power loading in order that it may be taken off safely and that it have a sufficiently rapid rate of climb to permit the clearing of obstacles. As an item of somewhat lesser immediate importance is the ability to attain desired cruising altitudes in a reasonable length of time. The net effect of these and other requirements has been to hold the land-transport airplane to a power loading upwards of 13 lb. per hp. available at takeoff. For example, Table I indicates the values typical of several widely used airplanes.

Table I—Power Loadings

Airplane	Power Loading (Gross Weight, Lb. per Hp. Available at Take-Off)
Boeing 247-D	12.4
Douglas DC-2	12.8
Douglas DC-3 (G-2 Cyclone)	12.0
Lockheed Electra	11.6
Douglas DC-4	13.5 (Approximate)

[This paper was presented at The National Aeronautic Meeting of the Society, Washington, D. C., March 11, 1937.]

[1] See S.A.E. TRANSACTIONS, August, 1936, pp. 304-312; "Future Possibilities of 100-Octane Aircraft-Engine Fuel," by Capt. F. D. Klein.

[2] See S.A.E. TRANSACTIONS, September, 1936, pp. 333-340; "A New High-Octane Blending Agent," by H. E. Buc and Major Edwin E. Aldrin.

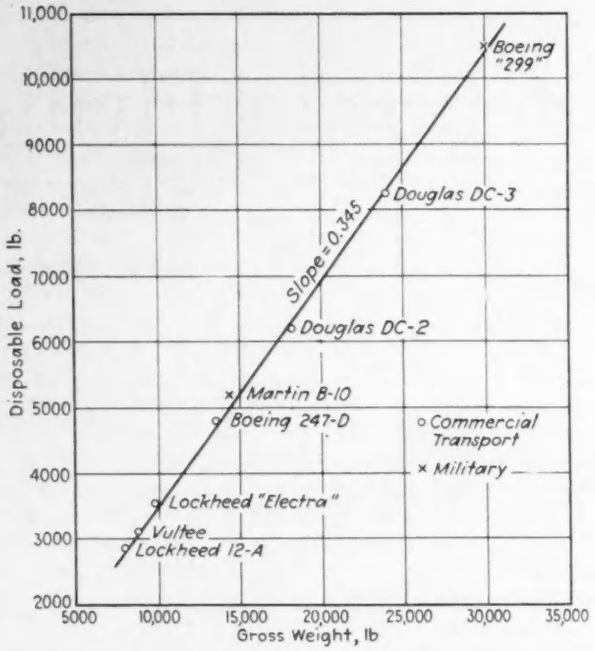


Fig. 1 – Disposable Loads of Current Land-Type and Land-Transport Airplanes (Plotted from Published Data)

The value 13 lb. per hp. is particularly applicable to several recent designs for the larger type of transport airplanes. It should be recognized that, if conditions permit the use of higher power loadings, however, larger loads can be carried with any given amount of available power as, for example, in the case of the long range transpacific type of flying boat now being operated by Pan American Airways.

Once the power loading has been set for any particular class of operation the total or "gross" weight which can be carried by any airplane is related directly to the horsepower available at take-off. This relation may be expressed very simply as follows:

$$\text{Gross Weight Permitted} = \text{Power loading} \times \text{Take-Off Horsepower}$$

$$\text{or} \quad W = w \times \text{B.Hp.}_m \qquad (1)$$

An examination of the load-carrying capacities of the various transport and long-range land-type airplanes that are being used currently indicates a very definite relation (for any particular type) between the useful or "disposable" load and the gross weight of the completely loaded airplane. Information which has been published is plotted in Fig. 1. It will be noted that, for the range of data indicated, the curve shows that 34.5 per cent of the gross weight of airplanes of this type is available in the form of disposable load. This relation may be expressed very simply as follows:

$$\text{Disposable Load} = \text{Gross Weight Permitted} - \text{Empty Weight}$$

$$\text{or} \quad U = 0.345\,W \qquad \text{(See Fig. 1)}$$

$$= 0.345\,w \times \text{B.Hp.}_m$$

$$= 4.5 \times \text{B.Hp.}_m \text{ where } w = 13 \text{ lb. per hp.} \qquad (2)$$

As the disposable load in an airplane must include the weights of fuel, lubricating oil, and crew necessary for a given flight, the net revenue or "payload" is represented by the disposable load less the combined weight of fuel, oil, and crew or:

$$R = U - (C + O + F)$$

$$= 4.5 \times \text{B.Hp.}_m - \left[C + O + \frac{D_c \times f_c \times \text{B.Hp.}_c}{Q \times S}\right] \qquad (3)$$

where D_c = Maximum Cruising Range, miles.

F = Weight of Fuel Required, lb.

f = Specific Fuel Consumption in lb. per b.hp-hr.

B.Hp._c = Cruising Horsepower Required.

S_c = Cruising Speed, m.p.h.

Q = Ratio of Heat Content of Fuel Used to That of Gasoline (20,500 B.t.u. per lb.).

C = Crew Weight, lb.

O = Weight of Engine Oil Carried, lb.[3]

It is obvious that the actual revenue load will vary between airplanes of the same type depending upon the actual size of the vehicle. However, a four-engine airplane weighing over 50,000 lb. gross may have exactly the same relative performance characteristics as a small twin engine airplane weighing less than a fifth as much. Regardless of the actual size of the engines used (within the range of current practice), or of the actual number of engines installed upon any particular airplane, the relative engine performance in terms of the number of horsepower hours which can be developed per unit of fuel used is fixed within very close limits. It becomes possible, therefore, to employ fuel *quantity* as a measure of the size of vehicle in question, and within the limits of current transport aircraft practice, one gallon of gasoline of any particular octane number will accomplish about the same amount of useful work regardless of the size of the airplane or the number of engines installed upon the ship in which the fuel is used. It then becomes possible to develop a simple expression for specific revenue, or net load-carrying capacity per gallon of gasoline used as follows:

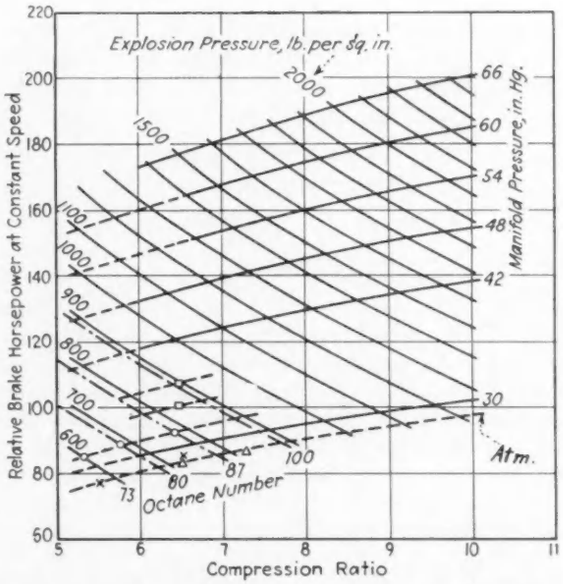


Fig. 2 – Calculated Explosion Pressures, Relative Outputs, and Boosts as Related to Compression Ratio and Fuel Octane-Number Requirements (Based on Fig. 10 from DuBois and Cronstedt[4])

[3] Strictly speaking, oil requirements will vary with consumption and time as is the case for fuel requirements. It is, however, necessary to maintain certain minimum levels in any system regardless of flight duration and, as this item is of quite minor import, it is more satisfactory to consider it a constant for any particular installation.

Specific revenue or net load capacity, based on 1 gal. of gasoline:

$$r_c = 6 \times \frac{\text{Revenue Load}}{\text{Fuel Weight}} \quad \begin{matrix}\text{(Approximate fuel density taken}\\ \text{at 6 lb. per gal.)}\end{matrix}$$

$$= 6\left[\frac{U - F - (C + O)}{F}\right]$$

or, for current conventional land transports:

$$= 6 \times \frac{\left[4.5\ \text{B.Hp.}_m - \dfrac{D_c \times f_c \times \text{B.Hp.}_c}{Q \times \omega_c} - (C + O)\right]}{\dfrac{D_c \times f_c \times \text{B.Hp.}_c}{Q \times \omega_c}}$$

$$= 6 \times \frac{\left[4.5 - \dfrac{D_c \times f_c \times \text{B.Hp.}_c}{Q \times S_c \times \text{B.Hp.}_m} - \dfrac{(C + O)}{\text{B.Hp.}_m}\right]}{\dfrac{D_c \times f_c \times \text{B.Hp.}_c}{Q \times S_c \times \text{B.Hp.}_m}} \qquad (4)$$

Calculated Effects of Octane Changes

At the 1937 Annual Meeting of the Society DuBois & Cronstedt[4] presented the results of a study of the effects of varying supercharge and compression ratio upon the relative power output and fuel economy of a hypothetical engine for a series of maximum combustion pressures. These pressures are, in turn, determined for any particular cylinder and piston set-up by the knocking characteristics of the fuel employed. Their theoretical studies have been supported by a carefully conducted experimental investigation carried out in the laboratory upon a special engine equipped with an aircraft-type cylinder. Fig. 2 has been taken from the DuBois & Cronstedt paper and is reproduced with slight modifications and additions. In addition to confirming the general relations the experimental work furnishes several "calibration" points for identifying the effects of octane-number changes. Further, although actual compression-ratio and supercharge values differ from those used in the prevailing large aircraft engines, the *relative* effects given agree almost exactly with those obtained in current practice. As an example the *relative* values for one particular design which is furnished for use on fuels of three different octane numbers, 73, 80 and 87, are shown in Fig. 2. From Fig. 2 it is possible to construct Fig. 3 which translates the theoretical relations between supercharge and compression ratio and power and air-cycle efficiency to relative output and economy as determind by octane number. Using values approximating those now being obtained in practice as a basis this chart furnishes all the engine information necessary to make possible the use of Equation (4). The relation assumed between cruising and take-off powers is purely empirical but is representative closely of general commercial practice. It should be noted that although A.S.T.M. or "Motor" method octane numbers have been used in this discussion, the general principles are applicable to any system of describing fuel-knocking tendencies *provided*, of course, ***that the method used measures this property in terms properly reflecting the service under consideration.*** It would probably be more in order at this time to use proportions of iso-octane and heptane rather than octane numbers by any particular system.

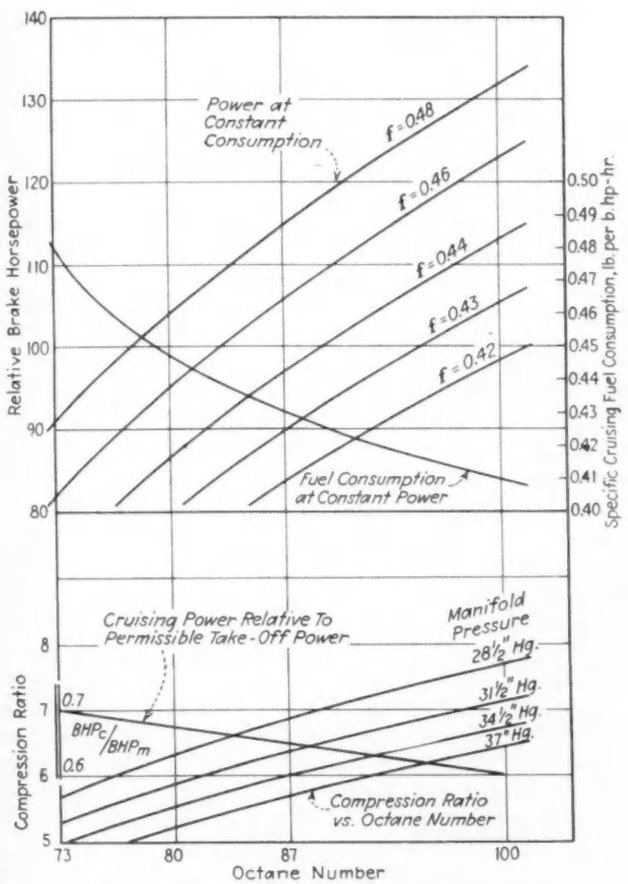


Fig. 3 – Relative Output and Economy as Determined by Octane Number

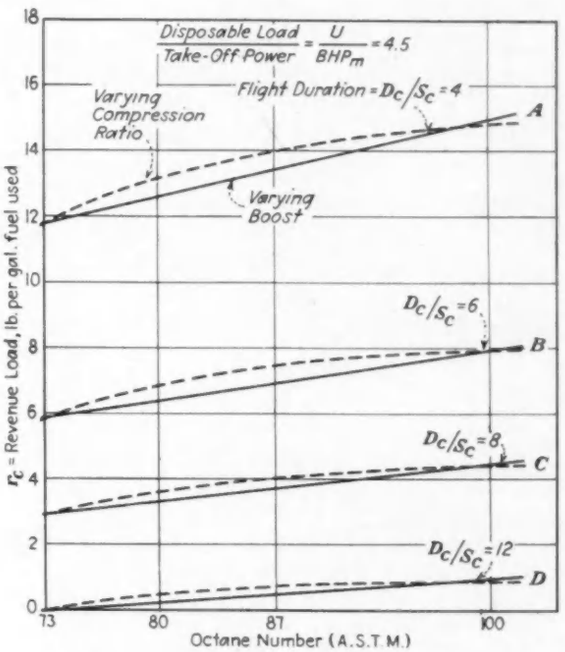


Fig. 4 – Revenue Load Vs. Octane Number

Several examples of inserting the engine-performance values of Fig. 3 in Equation (4) are shown in the accompanying Fig. 4. Curve *A* shows the revenue load in pounds which can be carried per gallon of fuel used for 4 hr. (maximum) in an airplane for which 4.5 lb. per take-off hp. are permitted. The solid line depicts the effect of utilizing increased octane numbers for securing higher take-off power (varying boost, only) whereas the dotted curve shows the effect of using these advantages to secure increased fuel economy (varying compression ratio, only). It will be noted that there is comparatively little difference between increasing payloads by the

[4] See S.A.E. TRANSACTIONS, June, 1937, pp. 225-231; "High Output in Aircraft Engines," by R. N. Du Bois and Val Cronstedt.

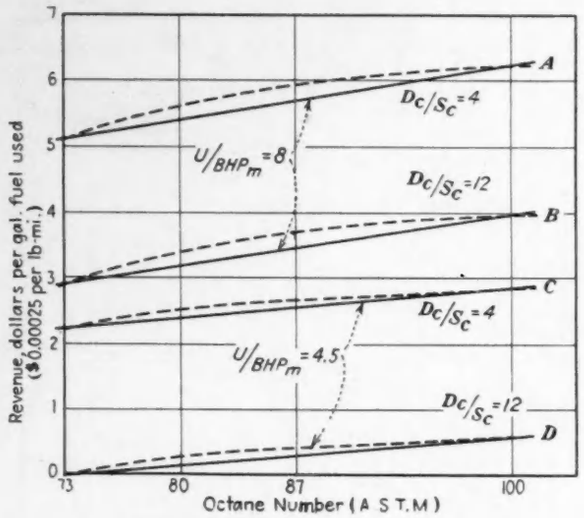


Fig. 5 – Representative Earning Capacities of Fuels of Varying Octane Number

supercharge and compression-ratio routes. For moderate octane-number increases the figures are favorable to the latter scheme. Very high compression ratios, however, apparently do not yield as attractive economic returns as supercharging to corresponding octane-number limits. Curves *B, C* and *D* in Fig. 4 show similar effects upon revenue loads for flight durations of 6, 8 and 12 hr. respectively.

In Fig. 5 are shown several combinations of disposable load (per take-off horsepower) and flight duration (D_c/S_c) in which the actual revenue earned in dollars per gallon of fuel used is plotted against fuel octane number. The basis for curves *A, B, C* and *D* is $0.00025 per lb-mile as representing (in round numbers) the revenue generally received for passenger transportation in the United States. This corresponds to 5 cents per passenger mile on the basis of 200 lb. for passenger and luggage carried free. The actual figure varies somewhat for certain flights and may not correspond even closely with mail and express revenues. However, it is felt that it may be used safely in the present study as it probably does agree well with *incremental* revenues which must, in general, come from passenger transportation. In any event, the actual values employed serve only to emphasize, and do not determine, the trends made possible by octane-number increases. Curve *B* in Fig. 5 indicates the approximate values of octane-number increases for an operation similar to that now in effect between San Francisco and Honolulu.

That the value of octane-number improvement increases with both increased payload ability of the ship (and its operating limitations) as well as with cruising range is shown in Fig. 6. In this figure the points marked *A* and *B* correspond to current land-transport and ocean-transport operations respectively. Although the increased earning capacity of an octane number may vary materially, it is emphasized clearly that such improvements are worth so much as to leave little doubt as to the direction octane-number trends must take. For example, current overland transportation evaluates an improvement of 1 octane number at 2¼ cents per gal. which, in turn, indicates that 1 gal. of 100-octane-number fuel can earn 29.3 cents more under these conditions than can 1 gal. of 87-octane-number fuel. Obviously such a profit cannot be realized simply by changing fuel in an existing airplane. These values apply only to *design changes* made possible by octane-number improvement. Further, increases in supercharge or compression ratio (particularly the latter) may result in greater maintenance costs, although in actual practice this penalty has not occurred. Probably the values shown in Fig. 6 should be penalized somewhat on this account; however, a 10 per cent penalty is certainly large enough.

The values given in Fig. 6 were based upon varying take-off power at a specific cruising fuel consumption constant at 0.48 lb. per b.hp-hr. If engine-design improvements should result in decreased consumptions not dependent on fuel octane number, then specific earning capacities will be affected as shown in Fig. 7. It is interesting to note that here, as in the previous figures, independent improvements in the airplane itself do not depreciate, but rather enhance, the value of any given fuel octane-number increase. Obviously, although octane-number changes are the principal subject of this paper, any other changes in fuel characteristics which may be reflected by engine-performance or fuel-consumption changes must produce similar effects. For example, changes in heat content produce inversely proportional effects on specific fuel consumption under any given set of conditions. Petroleum hydrocarbons provide the highest energy content on either a volume or a weight basis of any liquid fuel. For aviation gasolines this value corresponds to about 20,500 B.t.u. per lb.

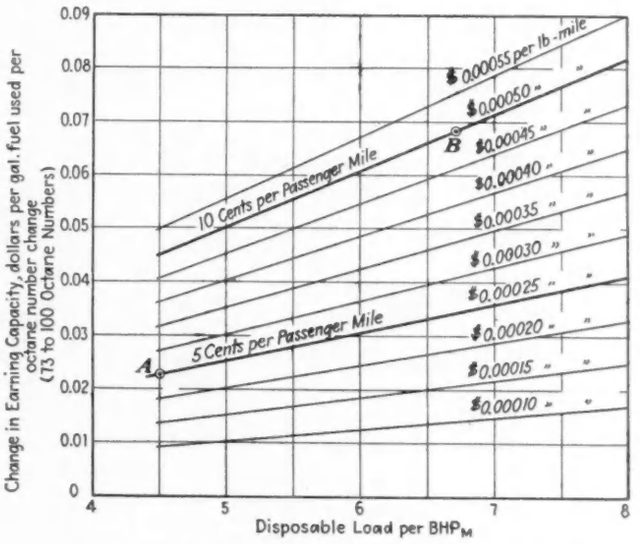


Fig. 6 – Change in Earning Capacity Per Gallon of Fuel Per Octane-Number Change

or about 123,000 B.t.u. per U. S. gal., these values varying slightly between individual gasolines. The use of non-petroleum fuels, however, particularly those containing oxygen (such as alcohols) introduces quite important reductions in heat content, the effects of which are illustrated in Fig. 8. Here it is indicated that a decrease of 1 per cent in heating value of the fuel must be accompanied by a loss in earning power corresponding to that of about 2 octane numbers. Due to the fact that reliable information on the performance characteristics of non-petroleum fuels is not available, it is impossible to evaluate such fuels completely at this time. In any case, however, it should be borne in mind that suitable allowances must be made for variations in heat content from that of gasoline.

Comments

The foregoing discussion is not intended as an exact computation of the earning power of any particular improvement in fuel knocking quality. Its purpose is merely to show "in which direction the wind must blow" in so far as design trends for the next few years are concerned. Also it must again be emphasized that, although higher octane-number fuels undoubtedly are justified, this conclusion applies pri-

marily to airplane and design. Rarely, if ever, will the opportunity come to increase the earning capacity of an existing airplane by an expedient as simple as merely pouring in higher antiknock fuel and opening the throttles wider. However, the fact that the advantages of improved fuel quality must be recognized in the design of the airplane and its powerplant in no way detracts from the fundamental worth of such changes.

In this study every effort has been made to be as conservative as possible in making claims for the earning power of octane-number improvements. Recently E. L. Bass[5] has presented a discussion of the utility of high-octane-number fuels which arrives at conclusions similar to the present, although expressed rather differently. In the Bass paper, however, the performance differences between previous engine designs for 87-octane-number fuels and new designs for 100-octane-number operations have been used as the basis for calculations. It would appear that some exception may be taken to this procedure on the ground that it credits to octane-number improvements the results of developments (such as better cooling) not directly affected or made possible by fuel quality changes. Such assumptions as made by Bass, however, only serve to emphasize the value of octane-number improvements and, if his basis is acceptable, the values indicated in this paper will be greatly increased.

The introduction of the heat-content factor may be regarded as unnecessary for most practical purposes as, in all probability, fuels having energy contents less than gasoline would be used for take-off purposes only when fuel consumption is relatively unimportant. It is felt, however, that such a picture would be a temporary one and that, as engine performances are increased, the time may well come when the exclusive use of high-octane-number fuels will be justified economically by making possible very low cruising fuel consumptions. Basically, therefore, it appears that oxygen-containing or hydrogen-deficient fuels and the like (particularly alcohol blends) must be penalized as indicated. This penalty would involve a decrease in earning power corresponding to a reduction of about two octane numbers for each per cent reduction in energy content. In the case of ethyl-alcohol blends this value would amount to a loss of the effect of two octane numbers for each three per cent of anhydrous alcohol.

No attempt has been made to assign military-performance values to fuel improvements. Obviously increases in octane number should be of great value for this class of service whether used to increase the performance of a combat plane or the range and capacity of a bomber. The conclusion that such improvements can be evaluated higher than for commercial service appears virtually inevitable.

It is also obvious that the advantages of octane-number improvements could be realized by increasing the speed of the airplane with attendant reductions in operating overhead and increased salability of service. Barnwell[6] illustrates the attendant problems rather aptly, and his article could be used as a very convenient basis for translating octane-number changes into possible speed changes. However, the means for evaluating such speed changes are not available to the present writer. Further, Barnwell's illustrations emphasize the unfavorable economics of increases achieved by the brute-power route for airplanes of present commercial type. Therefore, as significant speed increases must come as the result of refinements in airplane design, no attempt has been made to evaluate this factor in the present paper.

The values computed in this paper stop at 100 octane number. That this value does not constitute a true limit is emphasized by the work of DuBois & Cronstedt. Here it is shown that the addition of 3 cc. of lead tetraethyl per gal. to iso-octane results in a possible performance increase (above 100) corresponding to at least 30 A.S.T.M. octane numbers. It should be appreciated that A.S.T.M. octane numbers are not particularly well adapted to this problem as it is generally recognized that they do not truly reflect the characteristics of service engines. However, a thorough investigation of the aviation fuel-knock-testing problem is now in progress

[5] See *Aircraft Engineering*, January, 1937, pp. 13-18; "High Octane Fuels," by E. L. Bass.

[6] See the *Journal of the Royal Aeronautical Society*, Vol. 40, 1936, pp. 198-231; "Some Notes on Aircraft Possibilities," by Capt. F. S. Barnwell.

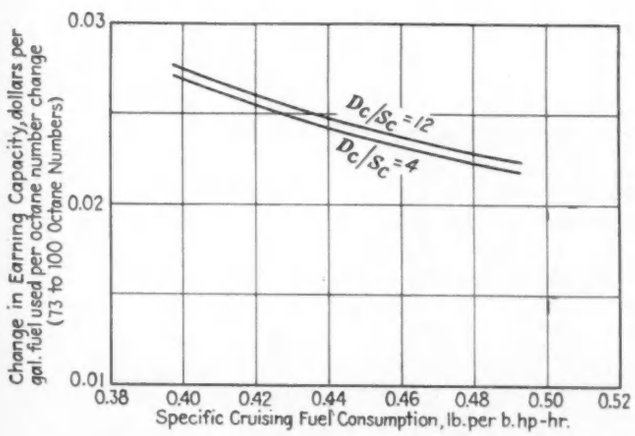


Fig. 7 – Change in Earning Capacity with Octane Number as Affected by Specific Fuel Consumption

(U/B.Hp.$_m$=4.5)

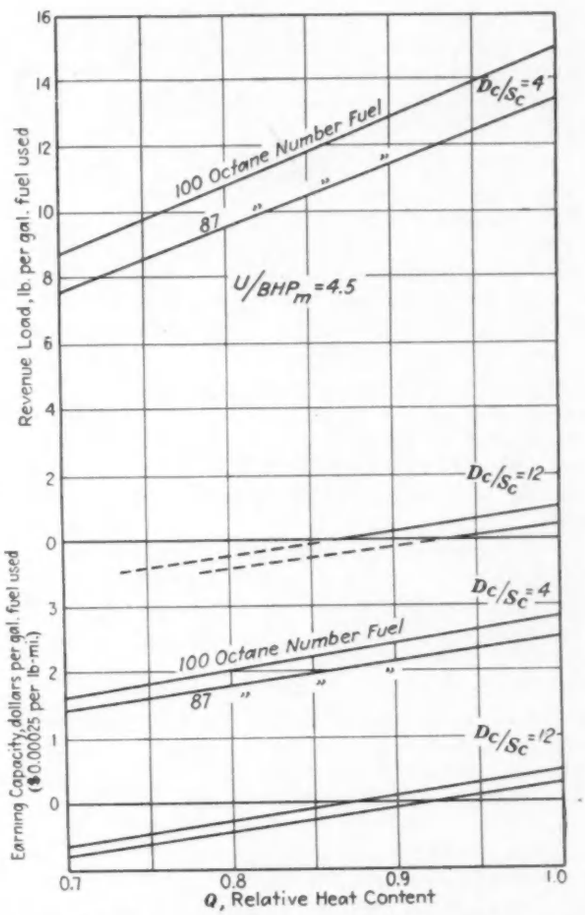


Fig. 8 – Effect of Heat-Content Decrease on Revenue Load and Earning Capacity

by the Aviation Fuels Committee of the C.F.R. When their work is completed, it should then be possible to repeat the present calculations on a more strictly quantitative basis.

Conclusions

It is felt that the foregoing study justifies the following conclusions:

(1) Octane-number improvements make possible advances in engine design and airplane carrying capacity of sufficient magnitude to determine further trends with a high degree of certainty.

(2) Depending upon type of vehicle and conditions of operation the value of an improvement of one octane number may range from slightly over 2 cents to 8 cents per gal.

(3) Independent improvements in airplane and engine performance serve to enhance the values of octane-number increases.

(4) For moderate octane-number increases, raising compression ratios seems to offer a slight economic advantage over supercharging. Above 100 octane number, a distinct improvement in earning power for increasing supercharge is indicated.

(5) Decreases in energy content of the fuel result in earning-capacity losses equivalent to about 2 octane numbers for each 1 per cent reduction in heating value.

(6) The foregoing conclusions apply only when the possible advantages are allowed for in design. It is not to be inferred that the earning power of an existing airplane can be increased materially by simply increasing the octane number of the fuel.

(7) Although no attempt has been made to include military service in this estimate, it appears that the value of octane-number increases must be even greater than for civil transport operation.

(8) The indicated increases in earning power are so much greater than any probable fuel-cost increases that the upward trend of octane-number requirements and use is clearly evident.

Acknowledgment

The writer is deeply grateful to the following friends: A. L. Beall, W. A. Parkins, Val Cronstedt, S. D. Heron, Opie Chenoweth, William Littlewood, H. O. West, and Capt. F. D. Klein. Their criticisms and advice have served to call attention (it is hoped) to the worst mistakes and to render the figures indicated more accurate than would be possible otherwise.

Appendix

Example of calculation of specific load and earning capacities based on 1 gal. of gasoline:

$$= 6 \times \frac{\left[U/\text{B.Hp.}_m - \dfrac{D_c \times f_c \times \text{B.Hp.}_c}{Q \times S_c \times \text{B.Hp.}_m} - \dfrac{(C+O)}{\text{B.Hp.}_m}\right]}{\dfrac{D_c \times f_c \times \text{B.Hp.}_c}{Q \times S_c \times \text{B.Hp.}_m}} \qquad (4)$$

(A) Assume:

U = 13 (lb. per B.Hp.$_m$) × 0.345 (Useful Portion of Gross Weight) = 4.5

D_c/S_c = Cruising time, hr. = 4.0

Fuel Octane No. = 73

$\dfrac{(C+O)}{Q}$/B.Hp.$_m$ = 0.5 = 1.0

From Fig. 3, B.Hp.$_c$/B.Hp.$_m$ = 0.7

$$\text{Therefore } r_c = 6 \times \frac{[4.5 - 4 \times 0.48 \times 0.7 - 0.5]}{4 \times 0.48 \times 0.7}$$

$$= 6 \times \frac{[4.5 - 1.35 - 0.5]}{1.35}$$

$$= 11.8$$

At 5 cents per passenger mile (at 200 lb.) 1 lb-mile = $0.00025 revenue.

Assuming S_c = 187.5 and D_c = 750

E_c = 750 × $0.00025 × 11.8 = $2.21

(B) Assume conditions same as (A) but fuel of 100 octane number, advantage to be taken by decreasing consumption and keeping maximum output constant. Then:

$$r_c = 6 \times \frac{[4.5 - 4 \times 0.41 \times 0.7 - 0.5]}{4 \times 0.41 \times 0.7}$$

$$= 6 \times \frac{[4.5 - 1.15 - 0.5]}{1.15}$$

$$= 14.9$$

$$E_e = 750 \times \$0.00025 \times 14.9 = \$2.80$$

(C) From (A) to (B)

E_c increases from $2.21 to $2.80 per gal.

Octane number, from 73 to 100

$$\text{Therefore, the increase in } E_e \text{ per octane number} = \frac{\$2.80 - \$2.21}{100 - 73}$$

= $0.022 per octane number

Factors Influencing Diesel Fuel-Injection Design

ORDINARILY, the fuel-injection equipment is considered as being a relatively delicate part of the Diesel engine. Our experience over the past six years, however, has demonstrated conclusively that, with reasonably simple precautions, injection equipment can be made to show life considerably in excess of that usually obtained from other parts of the machine. We do not hesitate to say that, barring accidents, the most prolific source of trouble in connection with premature failure of injection equipment is dirt in the fuel. Granting that dirty fuel is the principal cause of premature failure of the injection equipment, the precautions which can be taken to eliminate or minimize the danger arising from this source are worthy of discussion. Eliminating dirt from the fuel is most important, and yet quite simple to accomplish. We have accomplished this purpose: first, by providing suitable filters for cleaning the fuel before it reaches the injection equipment; second, by so designing the equipment that dirt will have the minimum effect; and, third, by providing easy means of replacement in case the system becomes damaged.

Since the fuel system must meet and cope with minute solid abrasives in the fuel in spite of all precautions to the contrary, the first consideration is the material from which the injection equipment must be made.

To resist the effect of minute solid abrasives which are able to pass filters of mesh as fine as 0.0012 in. and still wreak havoc with the metering and injection parts, demands material of high hardness. If it were possible to obtain or to produce suitable metals capable of resisting attrition, abrasion, or erosion from particles such as quartz sand charged with the high velocity of the fuel stream, then the problem would be solved easily.

Excerpt from the paper: "Some Factors Affecting the Design and Performance of Diesel Fuel-Injection Equipment," by J. M. Davies, E. W. Jackson, and G. C. Riegel, presented at the Tractor Meeting of the Society, Peoria, Ill., April 23, 1937.

Cab-Over-Engine Trucks – Their Place in Transportation

By Pierre Schon

General Motors Truck & Coach Division, Yellow Truck & Coach Mfg. Co.

WERE the original truck manufacturers on the right track thirty years ago when the "cab-over-engine" or, as called in those days, "motor-under-the-seat" type of design was considered standard practice?

Why was this type of design abandoned for the "engine-under-the-hood" type, and why was the revival of the cab-over-engine delayed or stifled until recent years?

Is the cab-over-engine truck design destined to take the place of the conventional type altogether, or will its adaptation be confined to a few vocational applications or a limited number of operators?

Has the wider usage of the cab-over-engine type been restricted due to mechanical limitations or has its higher cost, as compared with the conventional chassis of equivalent specifications, been the outstanding handicap, and to what extent will mass production of this type influence the shrinking of this price differential?

If the cost element differential could be eliminated, does there remain a place for the conventional engine-under-the-hood type in truck transportation?

Reasons why and wherefore, based on sound practical facts, are analyzed in this paper and, although the author refrains from making any definite predictions, nevertheless, it does not require a great deal of imagination to picture the cab-over-engine design taking a gradually more important place in truck transportation.

THE cab-over-engine type of truck is gradually taking a more prominent place in transportation. Manufacturers are fully aware of this trend and great strides have been made during the last few years in meeting the operators' demand for a type of vehicle which, in many ways, is the answer to some of the more important problems of modern transportation.

That the truck industry had its start with this type of vehicle, is evidenced by an extract from a 1911 catalog shown in Fig. 1 and it is interesting to note that the early builders did not give much consideration to that part of a motor truck which is of primary importance today, that is, cab design and driver comfort.

The advantages of this type of vehicle were recognized fully by the early builders, as evidenced by these statements from Fig. 1 dating back to 1911:

(1) Less overall length for a given length of loading space.

(2) Shorter wheelbase for a given length of loading space.

(3) Greater facility in handling in congested traffic and narrow streets.

(4) Less storage space required.

(5) Less space required at loading platforms.

(6) Less space required in street, whether vehicle is moving or standing.

(7) Better distribution of load.

The addition of three other important features brings the basic advantages of the cab-over-engine quite up to date.

(8) Greater safety – because of better visibility.

(9) Advantages in meeting legal size and weight restrictions.

(10) Attractive appearance and greater advertising value.

Why the C.O.E. Type Was Abandoned

With the exception of one American manufacturer, the production of the motor-under-the-seat or C.O.E. type was abandoned in this country, while European manufacturers continued the manufacture of the forward cab location consistently.

Appearance perhaps had a primary influence in the abandonment of this type of design in America for, as the operators demanded better protection for the driver, roll-up curtains with peep holes, enabling the driver to see where he was going, resulted in monstrosities as portrayed in Fig. 2.

The appearance of these vehicles may have been partly responsible for the laws in several states, requiring a motor vehicle to halt when approaching a horse and to remain stopped until the horse and wagon had passed by safely.

The failure of the early American truck builders to provide a more pleasing appearance in the front-end construction had perhaps a much greater influence in discontinuing this type of vehicle than any of the basic design principles. Advantages of shorter length, better load distribution, and so on, were of lesser importance for, at that time, there were no

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 5, 1937.]

drastic size and weight restrictions, no problems of load distribution due to legal axle-weight limitations. Solid tires were standard equipment, consequently no important tire-overloading nor spare-tire problems. Traffic congestion, as we know it today, did not exist and, therefore, some of the major inherent advantages of this design were disregarded by the operators. All these were contributory factors in favor of adopting the more graceful conventional design, which followed the constructional lines of the passenger car.

Perhaps the knockout blow was given when, in the Government specifications for War equipment, the forward location of the cab was eliminated entirely on all standard vehicles.

It is indeed a long step from the days of 1911 to 1937 and the place in transportation for the forward-cab location is being firmly reestablished, its streamlined appearance, as shown in Fig. 3, perhaps being a highly important inducement for accelerating public demand for this type of vehicle.

Although the pleasing appearance built into the C.O.E. front-end construction is a great spur for expansion of usage, nevertheless, the basic operating advantages of this design have become highly important in meeting the requirements of modern transportation from the legislative angle. No one can contest the fact that drastic reductions in size and weight regulations enacted in the various states during recent years have been partly responsible for the revival of this type of vehicle.

This factor may not seem so important when applied to the single vehicle; yet, where the legal length limit is only 26½ ft., as in Kentucky, the C.O.E. has a decided advantage over the conventional type as shown in Fig. 4. There is a gain of 200 cu. ft. possible as indicated in this illustration. To a moving-van operator, this additional load space is highly valuable for, on many moving jobs, the larger van body is capable of handling the entire load in one trip whereas, with 200 cu. ft. less, two trucks may be required to do the job. 20 states have enacted laws fixing the legal limit of a single vehicle below the 35 ft. length as recommended in the uniform code.

The advantages of the C.O.E. are quite pronounced in all other vocational operations where the type of commodity hauled requires maximum load space up to the limitations of the legal restrictions. This condition especially applies to vehicle combinations, and a typical example is found in the "truck-away haul" of new cars from the automobile factories as shown in Fig. 5. In this type of hauling, the economical load is four passenger cars and the minimum load-length requirement for the semi-trailer itself, is 33 ft., or a total minimum overall length of 40 ft. for tractor and semi-trailer. A load of only three cars, means a reduction of 25 per cent in the revenue per trip.

From every standpoint of safety and appearance, the tractor-semi has the advantage; yet the legal restrictions force the operators to use a contrivance like this Illinois Special. Almost all replacements in this particular vocation are now being made with C.O.E. tractors.

For a truck-and-trailer combination measuring 45 ft. overall, as recommended in the uniform code, the C.O.E. has the advantage of allowing two standard 18-ft. bodies and also a gain in additional payload as illustrated in Fig. 6. These increases are very important from the standpoint of earning capacity for commercial carriers. In eleven states, combination lengths have been reduced below the 45-ft. limit recommended in the uniform code. In several of these eleven states, the law does not actually prohibit the full trailer; yet, due to physical design and operating limitations, the restrictive length regulation makes the use of the trailer impractical with a truck of conventional design. In such cases, the C.O.E. not only has the advantage of more load space and more payload capacity in the truck itself, but the earning capacity is materially increased through the use of the trailer, without exceeding the legal length limitation.

This situation is illustrated clearly in the transportation of milk in the State of Illinois. The milk train shown in Fig. 7 complies with the Illinois length limitation of 40 ft., and the total gross weight of the combination is 72,000 lb. With a truck of conventional design, carrying a tank of the same size and capacity, there would be insufficient room to accommodate the four-wheel trailer. The trailer wheelbase could not be much shorter without endangering operating safety. It is true, a four-wheel truck with a smaller tank could be used, but that would mean a sacrifice of 16,000 lb. in the gross weight.

The C.O.E. makes it possible to utilize a six-wheel truck, loaded to the legal limit of 40,000 lb., plus a four-wheel trailer with a legal gross weight of 32,000 lb. – total G.V.W., 72,000 lb.

The three alternatives with the conventional type are:

(1) A single six-wheel truck – 40,000 lb. – loss in G.V.W., 32,000 lb.

(2) A four-wheel truck and four-wheel trailer – 56,000 lb. – loss in G.V.W., 16,000 lb.

(3) A tractor semi-trailer 40,000 lb. – loss in G.V.W., 32,000 lb.

Another interesting installation, shown in Fig. 8, brings out the advantages of the C.O.E. where a combination of vehicles is built up to the limit of the legal restrictions in load space and payload capacity. In this case the truck is of the four-wheel type, whereas the trailer is a six-wheeler. Built to operate in Wisconsin and Illinois, the 40-ft. length law applies, and the C.O.E. truck shows a decided advantage over the conventional type.

Although the C.O.E. vehicle costs more than the conventional unit, the slightly higher original cost is more than offset by the greater utility value as illustrated in Figs. 7 and 8. In both cases, there is a substantial increase in the gross weight, and the C.O.E. makes it possible to operate a more useful vehicle combination inside of the legal size and weight regulations.

Aside from the legal restrictive angle, the C.O.E. operated as a single vehicle or in a combination, has two inherent advantages of more load space and more payload capacity as shown in Fig. 9, the latter primarily resulting from a better load distribution in relation to the tire load capacity.

There is less danger of rear tires being overloaded, as the front tires carry their share of the gross weight, due to the 1/3 and 2/3 load distribution, a condition impossible to obtain with the conventional type.

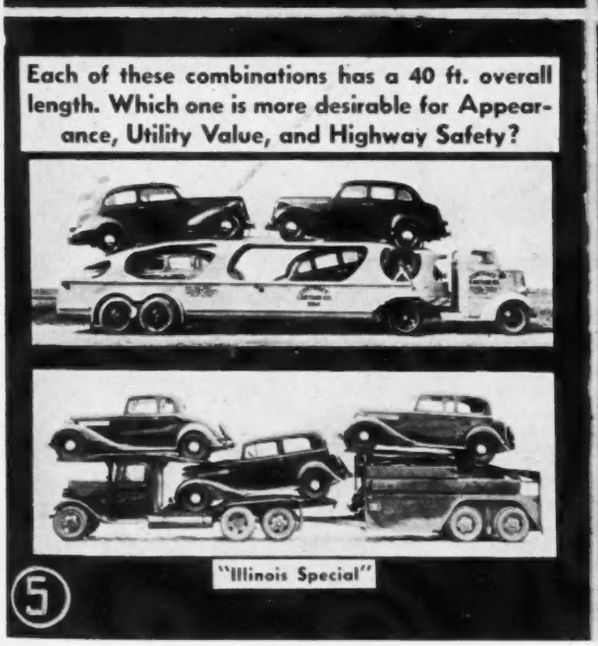
Note a gain of 2500 lb. in G.V.W., with the cab-over-engine, the rear axle being loaded to 15,000 lb. in both cases.

Fig. 10 clearly pictures the shorter length of the C.O.E. with a given load space. Note a gain of nearly 3 ft. in comparison with the conventional type, both in the single vehicle and also the tractor semi-trailer.

Aside from the C.O.E. advantages of more load space and greater payload, the shorter overall length requires less space at the loading platform. Note the location of the front wheels of the C.O.E. beer truck in Fig. 11 at the edge of the loading platform. The conventional type would extend several feet into the street or, in case this location would be prohibited by municipal ordinances, the body would have to be several feet shorter.

A large department store in Chicago three years ago installed a fleet of 8 C.O.E. tractors and 27 semi-trailers. The C.O.E. tractor made it possible to replace the 16-ft. semi-

Summary of Advantages Possessed by the "Motor Under the Seat" Type of Truck
1911
Less overall length, for a given length of loading space.
Shorter wheel-base for a given length of loading space.
Greater facility in handling in congested traffic and narrow streets.
Less storage space required.
Less space required at loading platforms.
Less space required in street, whether vehicle is moving or standing.
Better distribution of load
Reliance
Rapid
1


Each of these combinations has a 40 ft. overall length. Which one is more desirable for Appearance, Utility Value, and Highway Safety?
"Illinois Special"
5

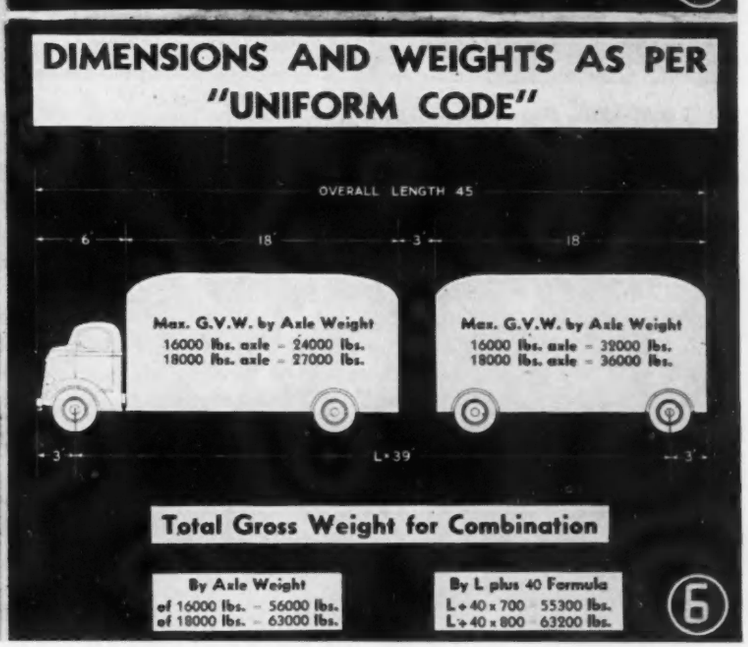


Fig. 1 – 1911 Vehicles

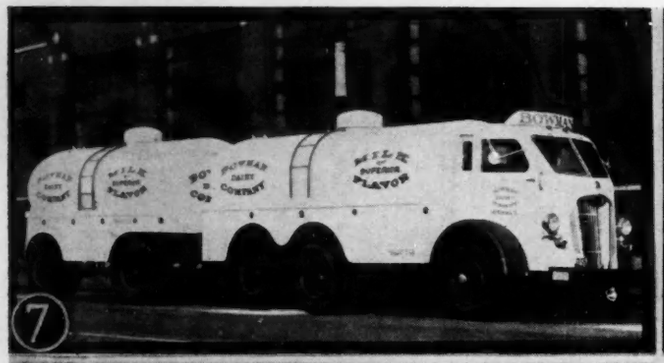
Fig. 2 – 1911 Driver Enclosure

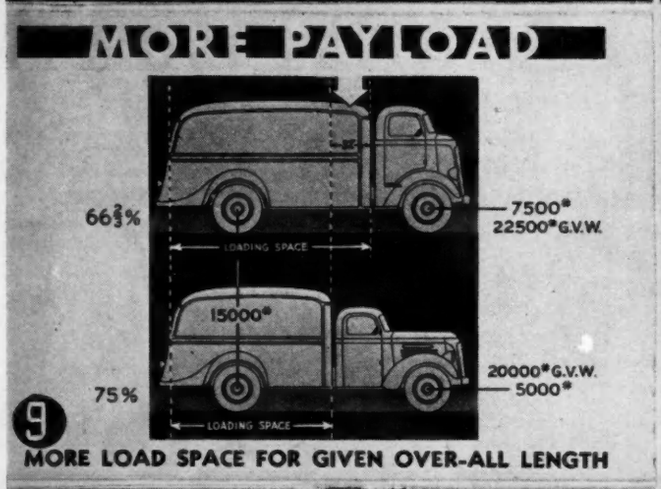
Fig. 3 – 1937 C.O.E. Designs

Fig. 4 – 26½-Ft. Moving Van

Fig. 5 – Comparison of Truck-Away-Haul Combinations

Fig. 6 – Chart for Truck-and-Trailer Combination





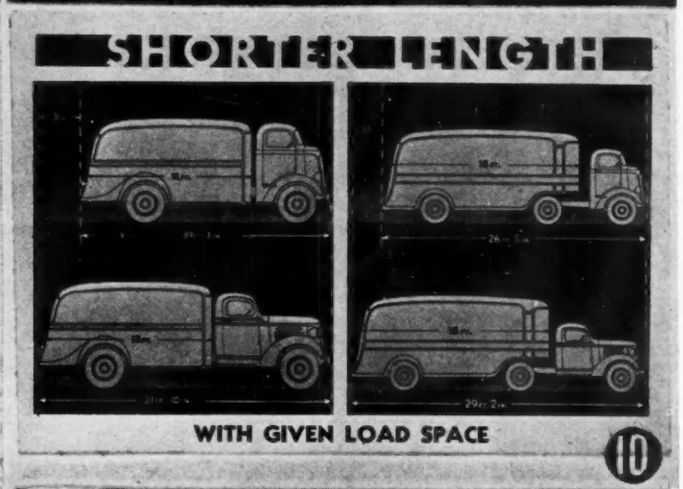


Fig. 7 – Milk Train

Fig. 8 – C.O.E. Truck-Trailer Unit Illustrating Load Space and Payload Capacity

Fig. 9 – More-Payload Chart

Fig. 10 – Shorter-Length Chart

Fig. 11 – C.O.E. Beer Truck at Loading Platform

Fig. 12 – Department Store C.O.E. Tractor-Semi-Trailer

trailers previously used, with a new 22-ft. semi-trailer, a gain of 6 ft. in the loading space of each trailer body. Fig. 12 shows one of these C.O.E. units.

The narrow space available between two loading docks allowed barely sufficient room to maneuver a conventional-type tractor with a 16-ft. semi-trailer. Bulky merchandise made it necessary to use longer trailer bodies. Building alterations were out of the question, due to prohibitive costs. The C.O.E. tractor solved this problem and the 22-ft. trailer bodies are being handled with much greater ease, due to the excellent maneuverability of the 96-in. wheelbase C.O.E. tractor as compared with the conventional type formerly used which had a 150-in. wheelbase. In this case there was an actual gain of 6 ft. in the body length, primarily due to the easier maneuverability and shorter length of the C.O.E. tractor.

Truck operators and fleet superintendents do not always realize the advantages in the time-saving element at loading docks and in congested traffic brought about by the easier handling and the shorter turning radius of the C.O.E. vehicle, illustrated in Fig. 13. There is nothing more enlightening than a conversation with a driver who previously handled the conventional type of vehicle in heavy traffic. Place him on a C.O.E. tractor, especially one of recent design, and you will find invariably, that the driver is a great booster for this type of vehicle.

From the standpoint of highway safety, the C.O.E. also takes first prize in the contest for road visibility as shown graphically in Fig. 14. Taking two trucks of the same capacity with the same size front tires, we find a difference of 15 ft. measured from the driver's eyes to the center of the road ahead.

A trend toward streamlined appearance during recent years, has affected road visibility on the conventional type, due to higher radiators, longer hoods, and a lower position of the driver's seat. The efforts on the part of the designers to give the 1½-ton truck the appearance of a 3-tonner have not been contributory factors to highway safety, resulting in better visibility.

Highway safety factors are built into the C.O.E. type due to the forward location of the driver. Inspection of Fig. 15 shows that there is less danger of exhaust gases entering the cab. The C.O.E. provides entrance of clean air; it eliminates excessive heat at the floor boards in hot weather; and all metal parts, such as dashboard, pedals, and so on, are cooler in the C.O.E. than in the conventional type due to their forward location and the quicker dissipation of heat from the engine toward the rear back of the cab, instead of passing out under the floor boards as in the conventional type.

Perhaps one of the most outstanding examples of increase in utility value is accomplished through an application of the C.O.E. principle in the conversion of a standard 112-in. wheelbase ½-ton delivery chassis and a special 257 cu. ft. body for an operation where body load space is of greater importance than payload capacity, as shown in Fig. 16.

Load-space dimensions are: length 108 in., width 68 in., and height 62 in.

In installations where light bulky merchandise is handled and large body load space is of utmost importance, the C.O.E. type also has advantages from the standpoint of original investment, when figured on the basis of *cost per cubic foot* of load space.

In the comparison shown in Fig. 17, we have taken a standard factory-built panel body with a load space of 135 cu. ft., against a special-built 257 cu. ft. body, incorporating the C.O.E. principle. The cost per cubic foot for the C.O.E. type conversion and the body, is indicated as three different cost figures, as bodies of this type are not available in standard factory production and there is a wide variance in the price of custom-built bodies. It is interesting to note that, even with an assumed list price of $700.00 for the conversion and the custom-built body, against a factory cost of only $229.50 for the standard panel body, the C.O.E. type still retains a price advantage of 37 cents per cu. ft., that is, $4.25 against $4.62. With the assumed lower cost prices of $500 or $600 for the custom-built bodies, the C.O.E. type has still a greater advantage from the investment standpoint, especially in those vocations where large load space is essential for handling light bulky merchandise and where the cost per cubic foot of body load space receives consideration in the purchase of new equipment.

Cost Per Cubic Foot of Body Load Space

	Conversion Unit and Special Body			Standard Panel
Chassis F.O.B. Factory	$395	$395	$395	$395
Body List Price	*500	*600	*700	230
Total List	895	995	1095	625
Body Cu. Ft.	257	257	257	135
Cost per Cu. Ft.	$3.48	$3.87	$4.25	$4.62

*Estimated.

Although the cost per cubic foot of the C.O.E. type, as shown in this comparison, is less than that of the standard body unit, nevertheless, the higher original investment for the C.O.E. custom-built body is a serious handicap and will remain so until this type of delivery truck is made available in quantity production as a standard factory-built model, complete with the unit type of panel body. There is a definite place in transportation for the C.O.E. type in the light delivery field, and there is a fast-growing demand for a light delivery truck which can be parked at the curb in the same space as can a passenger car.

In the retail delivery of merchandise, the parking problem has become so serious in our metropolitan areas that merchants, in some cases, have been forced to the two-man operation of a delivery truck. Double-parking being prohibited by police regulations, the C.O.E. has a decided advantage for parking at the curb, considering that a 1½-ton truck of this type with a 9-ft. body, such as shown in Fig. 18 has a shorter wheelbase and overall length than the average passenger car. In retail delivery where every stop involves finding a suitable parking space at the curb, substantial economies have been obtained with these shorter vehicles through time savings and increase in deliveries per working day.

The various operating advantages of the C.O.E. type in heavy traffic played a very important part in setting up the specifications for 315 new garbage collection trucks, recently installed by the New York City Sanitation Department. As shown in Fig. 19 the specially designed chassis is equipped with a 24 cu. yd. body, indicating that maximum obtainable load space was one of the deciding factors in favor of the C.O.E. type. The shorter, and therefore sturdier, frame is highly important in the dumping operation of this huge body. In this particular operation, where heavy traffic congestion is a serious problem, the conventional truck has definitely lost its "place in transportation."

In bus transportation, the operator's demand for additional seating capacity already has obsoleted the engine-under-the-hood type of design. In the process of style change during recent years, the first departure from the so-called conventional design practice consisted of placing the driver's seat in a forward location alongside of the engine. However, bus design has advanced a step further, by locating the complete powerplant at the extreme rear end of the vehicle, thereby utilizing the utmost maximum available load space for passenger seating capacity.

The "rear-engine" coach, apparently is destined to become the accepted standard in the bus industry for, in this phase of highway transport, there is a very definite "handwriting on the wall" that the "place in transportation" for the so-called "conventional type," will be extremely limited.

In the hauling of merchandise the great variety of vocational applications precludes the possibility of rear-engine location in motor-truck chassis. This condition especially applies to tractor and dump-truck operations where such a change in design would be decidedly impractical. However, the cab-over-engine principle can be used to an advantage for any and all types of vocational applications.

What are the obstacles holding back the expansion of usage of the C.O.E. type of truck?

Up to 1937 the comeback of the C.O.E. type was handi-

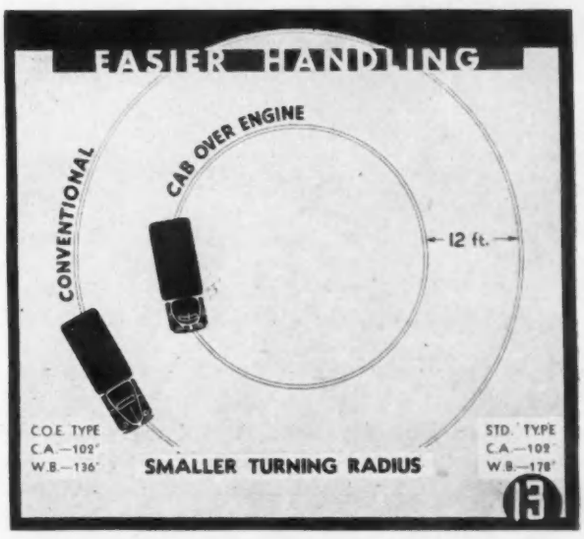
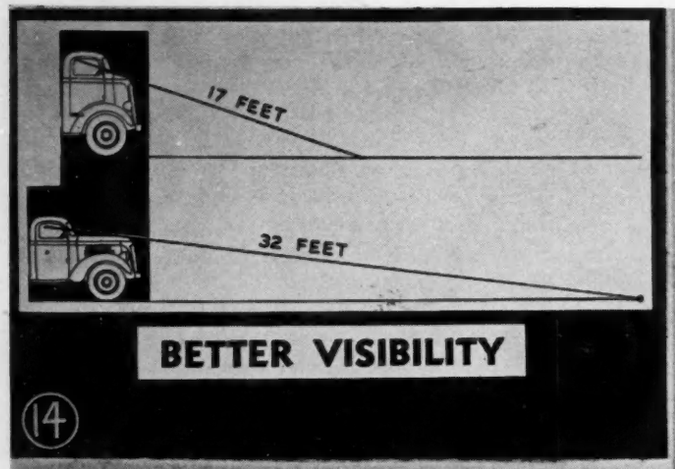
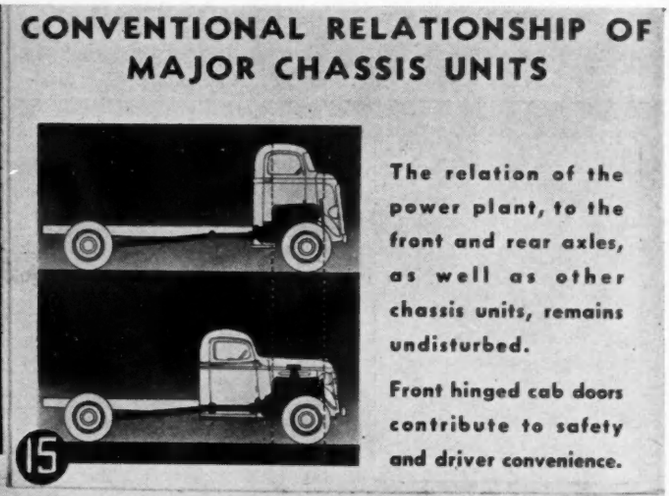


Fig. 13 – Easier-Handling Chart




CONVENTIONAL RELATIONSHIP OF MAJOR CHASSIS UNITS
The relation of the power plant, to the front and rear axles, as well as other chassis units, remains undisturbed.
Front hinged cab doors contribute to safety and driver convenience.
15

City of New York Department of Sanitation
24-Cubic Yard Dump Body
Load Space Length - 17'—9"
" " Width - - 7'—2"
" " Height - - 5'—3"
Wheel Base - - - 170"
Total Gross Weight, 40000 lbs.
19

Fig. 14 – Better-Visibility Chart

Fig. 15 – Conventional Relationship of Major Chassis Units

Fig. 16 – ½-Ton Conversion

Fig. 17 – C.O.E. and Conventional Models Compared on Cost Basis

Fig. 18 – C.O.E. 1½-Ton Delivery Truck

Fig. 19 – C.O.E. Garbage-Collection Truck

capped due to limited production and excessively higher prices in comparison to the conventional type. However, several of the larger truck manufacturers have entered this field with light-duty models and a more complete line, ranging from the 1½-ton size up to the largest capacities allowed by our state laws. These ultra-modern vehicles are now available at a much lower cost differential, primarily due to a larger production volume.

A listing of models produced by 8 major manufacturers of the C.O.E. type is shown in Table 1; it indicates that the manufacturers have utmost faith in the potential sales volume for this type of vehicle. The various models are grouped

Table 1 – C.O.E. Models – List Prices of Chassis and Cab F.O.B. Factory as of March 31, 1937

G.V.W. Ratings, lb.	4,000 7,000	10,000 12,500	12,501 15,000	15,001 17,500	17,501 20,000	20,001 22,500	22,501 25,000	25,001 27,500	27,501 30,000	30,001 36,000	1937 Total
Autocar	.....	.....	.....	.....	.....	.....	.....	$4,750	$5,350	$4,200	
	.....	.....	.....	.....	.....	.....	$4,150	4,900	5,750	5,000	11
	.....	.....	.....	.....	.....	.....	.....	5,950	6,950	6,200	
	.....	.....	.....	.....	.....	.....	.....	.....	.....	6,250	
Federal	.....	$1,250	$1,595	$1,785	$2,220	.....	.....	.....	.....	.....	4
GMC	.....	.....	970	.....	.....	.....	.....	.....	.....	.....	
	.....	830	1,165	1,665	1,895	$2,160	2,560	2,860	4,360	4,550	11
	.....	.....	1,290	.....	.....	.....	.....	.....	.....	.....	
IHC	.....	995	.....	.....	.....	.....	.....	.....	.....	.....	1
Mack	.....	.....	1,295	.....	.....	.....	.....	.....	.....	.....	
	.....	.....	1,595	.....	3,600	3,775	4,250	5,150	.....	6,100	7
Reo	.....	.....	1,425	.....	.....	.....	.....	.....	.....	.....	1
Studebaker	.....	850	1,155	1,705	2,095	.....	.....	.....	.....	.....	4
White	.....	.....	2,075	2,600	2,875	3,400	3,780	6,000	.....	6,825	7
Totals	.....	4	9	4	5	3	4	6	4	7	46
Average List Price	.....	$981	$1,396	$1,939	$2,537	$3,112	$3,685	$4,935	$5,602	$5,589	

according to gross vehicle weight and also by makes. These 8 manufacturers are offering a total of 46 models ranging in gross weight from 10,000 to 36,000 lb. Note that in the G.V.W. classification of 4,000 to 7,000 lb., that is, the light delivery bracket, no standard factory built models are available. So far the demand for the C.O.E. type in the package delivery size is met by various local body builders who also convert the chassis by changing location of steering gear and controls.

Whatever its proper designation may be, cab-over-engine – cab-forward – engine-under-seat – engine-between-seats – underslung powerplant – camel-back – and so on, the availability of this type of vehicle through *mass production* instead of being *custom-built,* has already brought about a shrinkage of the price differential in the 1937 models. Should this trend toward a lower price differential continue through larger production schedules of the C.O.E. type, it is quite possible that its many operating advantages and greater utility value will make a place for the cab-over-engine in transportation to the extent where the question may arise in future years:

Is there a place in transportation for the conventional type of truck?

The Determination of Ratings for Transport Aircraft Engines

(*Continued from page* 404)

countered in a much longer period under the same conditions and, by the same token, the test proves a large factor of safety at cruising output stress levels. This point is clear from an examination of a curve for steel showing the relation between number of stress reversals and unit stresses for a probable fatigue failure. In the case of aluminum, the same margin of safety is not obviously secured by the test procedure but, since the program includes 300 hr. of operation (approximately 18,000,000 stress reversals) at stress levels equal to or greater than stresses during cruising operation, the factor of safety appears adequate.

No feasible testing program reasonably can be expected to eliminate the need for extensive experience with an engine in routine flight service because of the limited number of test units involved and on account of other unavoidable limitations in laboratory test procedure.

The engine rating and durability tests which have been described are suggested for consideration as a part of a continuing program for increasing the safety of air transport, and to promote a uniformly high standard of performance for American aircraft engines. Adoption of this or some similar method of establishing engine ratings involves some temporary hardships in meeting a competitor who follows any less stringent practice. However, experience seems to demonstrate that a high standard of performance is the best insurance of success in a continuing struggle. The achievements of diligent technical research already have given the American aircraft industry no mean advantage in the competition for supremacy in the field of long-range transport, and it is essential that high standards be adopted in such matters as determination of engine ratings in order that unfortunate mistakes or wilful errors of a single competitor may not be permitted to jeopardize the interests of the whole industry.

Vibration Measurement in Flight

By C. S. Draper, G. P. Bentley, and H. H. Willis

Massachusetts Institute of Technology *Sperry Gyroscope Co., Inc.*

EQUIPMENT for measuring vibration in airplane structures and powerplants during actual flight is described in this paper. This development is the result of a cooperative research program carried out by the Bureau of Aeronautics of the U. S. Navy and the Massachusetts Institute of Technology with contributions of improvements in design and new features by the Sperry Gyroscope Co., Inc.

In its essentials, the M.I.T.-Sperry Apparatus consists of a number of electrical pickup units which operate a central amplifying and recording unit. The recorder is a double-element photographic oscillograph. Each pickup is adapted especially to the type of vibration that it is intended to measure and is made so small that it does not appreciably affect the vibration characteristics of the member to which it is attached rigidly. By using a number of systematically placed pickups, all the necessary vibration information on an airplane can be recorded during a few short flights. The paper takes up in detail flight test installation, sample records and results from flight test, measurement of vibratory strains, pickup units, strain gage, amplifier, oscillograph, and calibrator.

VIBRATION control in aircraft has become an increasingly important subject during the last few years. Not only has the primary problem of mechanical failures been attacked, but the equally serious matter of relieving passengers and crew from fatigue due to vibration has received much attention.[1,2,3,4,5] Experience has shown that a satisfactory solution to vibration difficulties can only be obtained if quantitative information on the conditions to be remedied is available. These data can be most readily obtained by the use of instruments specially adapted to the problem. In order to develop equipment suitable for measuring vibration in airplane structures and powerplants during actual flight,[6] a cooperative research program has been carried out by the Bureau of Aeronautics of the United States Navy and the Massachusetts Institute of Technology. The Sperry Gyroscope Co., Inc., became interested in the work and has contributed both improvements in design and new items which extend the capabilities of the apparatus.

In its essentials, the M.I.T.-Sperry Apparatus[7] consists of a number of electrical pickup units which operate a central amplifying and recording unit. The recorder is a double-element photographic oscillograph. Each pickup is adapted especially to the type of vibration that it is intended to measure and is made so small that it does not appreciably affect the vibration characteristics of the member to which it is rigidly attached. By using a number of systematically placed pickups, all the necessary vibration information on an airplane can be recorded during a few short flights.

Flight-Test Installation

Fig. 1 shows an airplane installation for a survey of structural and powerplant vibration. Typical pickup locations are shown on the engine mount, on the tail, and on the wing strut. A pair of wires from each pickup are carried to the distribution panel which is connected to the control panel by a multiple-conductor cable. A tap switch on the control panel is used to connect any two pickups to the amplifiers mounted in the baggage compartment. The records are made by the recording oscillograph placed in front of the operator. In other installations where space is limited the oscillograph can be placed in a compartment away from the cockpit and operated by an electrical remote-control system.

For a complete analysis of vibration into its components one pickup is required for each degree of freedom. A typical arrangement of pickups is shown in Fig. 2. The units mea-

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 11, 1937.]

[1] See S.A.E. TRANSACTIONS, October, 1931, pp. 513-530; "A Study of Airplane and Instrument-Board Vibration," and November, 1932, pp. 445-456; "Vibration of Instrument Boards and Airplane Structure," both by Stephen J. Zand.

[2] See the *Journal of the Aeronautical Sciences*, June, 1937, pp. 316-319; "Aircraft Vibration Flight Testing," by G. P. Bentley.

[3] See Dossier d'insonorarisation du Bregeut – Wibault 670 – "Etude du comfort du avion de transport et application pratique a un appareil," and *Journal of the Royal Aeronautical Society*, July, 1936, pp. 524-556; "Comfort in Air Travel," both by Stephen J. Zand.

[4] See the *General Electric Review*, September, 1934, pp. 423-428; "The Transient Vibrations of Machines," by W. E. Johnson.

[5] See Research Bulletin No. 44, Engineering Experiment Station, Purdue University, May, 1933; "Riding Comfort Analysis," by H. M. Jacklin and G. J. Liddell.

[6] See the *Journal of the Aeronautical Sciences*, February, 1936, pp. 116-121; "Measurement of Aircraft Vibration During Flight," by G. P. Bentley and H. H. Willis.

[7] See the *Journal of the Aeronautical Sciences*, May, 1937, pp. 281-285; "M.I.T.-Sperry Apparatus for Measuring Vibration," by C. S. Draper, G. P. Bentley, and H. H. Willis.

sure the vibration components indicated in the following table:

Complete Engine Motion	(1) Vertical engine vibration – mean of motion of Units 1 and 2.
	(2) Fore-and-aft engine vibration – mean of the motion of Units 3 and 4.
	(3) Lateral engine vibration – mean of the motion of Units 5 and 6.
	(4) Torsional crankcase vibration about span axis – proportional to the difference of motion between Units 1 and 2.
	(5) Torsional crankcase vibration about vertical axis – proportional to the difference of motion between Units 3 and 4.
	(6) Torsional crankcase vibration about thrust axis – proportional to the difference of motion between Units 5 and 6.
Crankshaft Vibration	(7) Torsional vibration of crankshaft – Unit 7.
Structural Vibration	(8) Vertical vibration of cabane struts and firewall – mean of motion of Units 8 and 9.
	(9) Torsional vibration of firewall section – proportional to difference of motion between Units 8 and 9.
	(10) Vertical shear distortion in engine mount structure (approximate) – motions of Units 1 and 2 combined with Units 8 and 9.
	(11) Torsional distortion in engine-mount structure (approximate) – motion of Units 5 and 6 combined with Units 8 and 9.
	(12) Vertical vibration of tail assembly at root of stabilizer – Unit 10 (see Fig. 1).
	(13) Vertical vibration at interplane strut – Unit 11 (see Fig. 1).

Fig. 3 shows the distribution panel attached to a motor-mount member. The individual wires are clamped to binding posts at *E* and a multiple-conductor cable is wired to the plug, *A*. Two pickups are visible in the photograph. In Fig. 4 is shown the control panel *B* with the tap switch, *C*. The leads, *D*, are connected to the shock-mounted amplifiers of Fig. 5 which, in turn, are connected to the oscillograph recorder.

In order to phase the records with respect to the engine crankshaft position an insulated wire is wrapped around an unshielded spark-plug wire of the master cylinder and connected to one of the amplifiers through a high resistance and a small condenser. With this arrangement, a distinctive mark is left on the vibration record each time that the cylinder fires.

With the arrangement described previously, records from six pairs of pickups can be taken within 30 sec., and the data for a complete analysis require about 1 hr. of flying time.

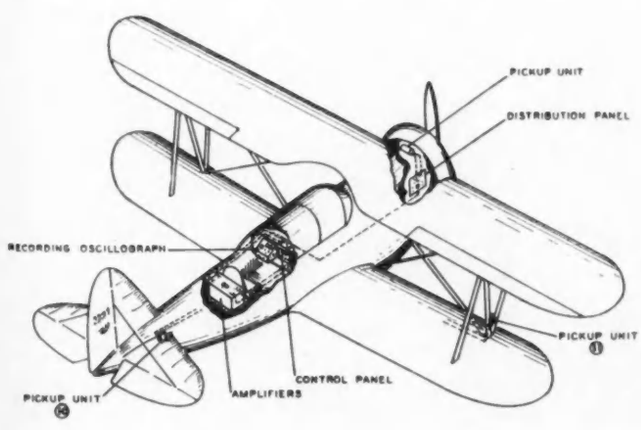


Fig. 1 – Installation of Vibration-Measuring Equipment for Flight Tests

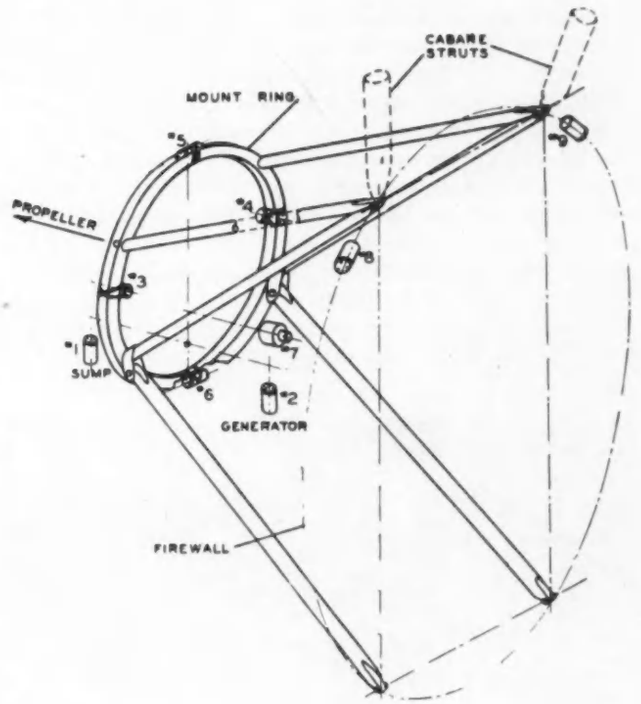


Fig. 2 – Location of Pickup Units for Analysis of Engine and Airplane Structural Vibration in Flight

Sample Records and Flight-Test Results

The traces shown at *a* in Fig. 6 are typical records from a pair of pickups placed in positions 5 and 6 of Fig. 2. With such an arrangement, displacements which are in phase at the two units represent linear vibration parallel to the wing span, whereas out-of-phase displacements mean torsional vibration of the engine crankcase about the longitudinal axis of the airplane. The linear and torsional components of vibration can be determined as curves formed by algebraic addition and subtraction of the original records according to the formulas (1) and (2):

$$X = \frac{xk_5 + yk_6}{2} \qquad (1)$$

$$\Theta = \frac{xk_5 - yk_6}{d} \qquad (2)$$

x = displacement on record from pickup 5.
y = displacement on record from pickup 6.
d = distance between pickups.
X = displacement of a point halfway between the pickups due to linear component of vibration.
Θ = angular displacement due to torsional vibration component.
k_5 and k_6 = calibration constants of pickups 5 and 6.

The curves *b* and *c* of Fig. 6 were determined from the records *a*, by the addition and subtraction processes as outlined

previously. For the particular conditions used in this test, the results show a linear vibration of the engine from side to side at twice engine-revolution frequency and a torsional vibration at engine frequency. In the usual case, an exhaustive analysis to determine the higher order components is not carried out unless special problems justify the effort.

A complete vibration study will consist of records made at a series of engine and airplane speeds to cover the operating range. The individual records are then broken down into components, and the amplitudes thus determined are plotted against frequency in the form of "resonance curves." Fig. 7 shows a set of such amplitude curves giving the vibration of an engine crankcase in torsion about the thrust axis and in translation along the wing-span axis. The pronounced engine revolution frequency peak at 1550 r.p.m. is accompanied by a decrease in the second order translational vibration. Curves of vertical vibration for the wing strut and the stabilizer taken simultaneously with the records of Fig. 7 are shown in Fig. 8. The first-order resonance peaks in strut and stabilizer vibration which occur at 1550 r.p.m. show that the engine-mount resonance at this speed is accompanied by a strong shaking of the whole airplane. There is also a resonance frequency in the stabilizer of approximately 4800 cycles per min. This resonance frequency is slightly excited at low engine speeds by the fourth-order vibration from the engine and appears more strongly when the second-order engine vibration approaches the resonant frequency.

The results just described show that the test airplane is subject to vibration due to an improperly constructed engine mount with a resonance peak just below the operating range

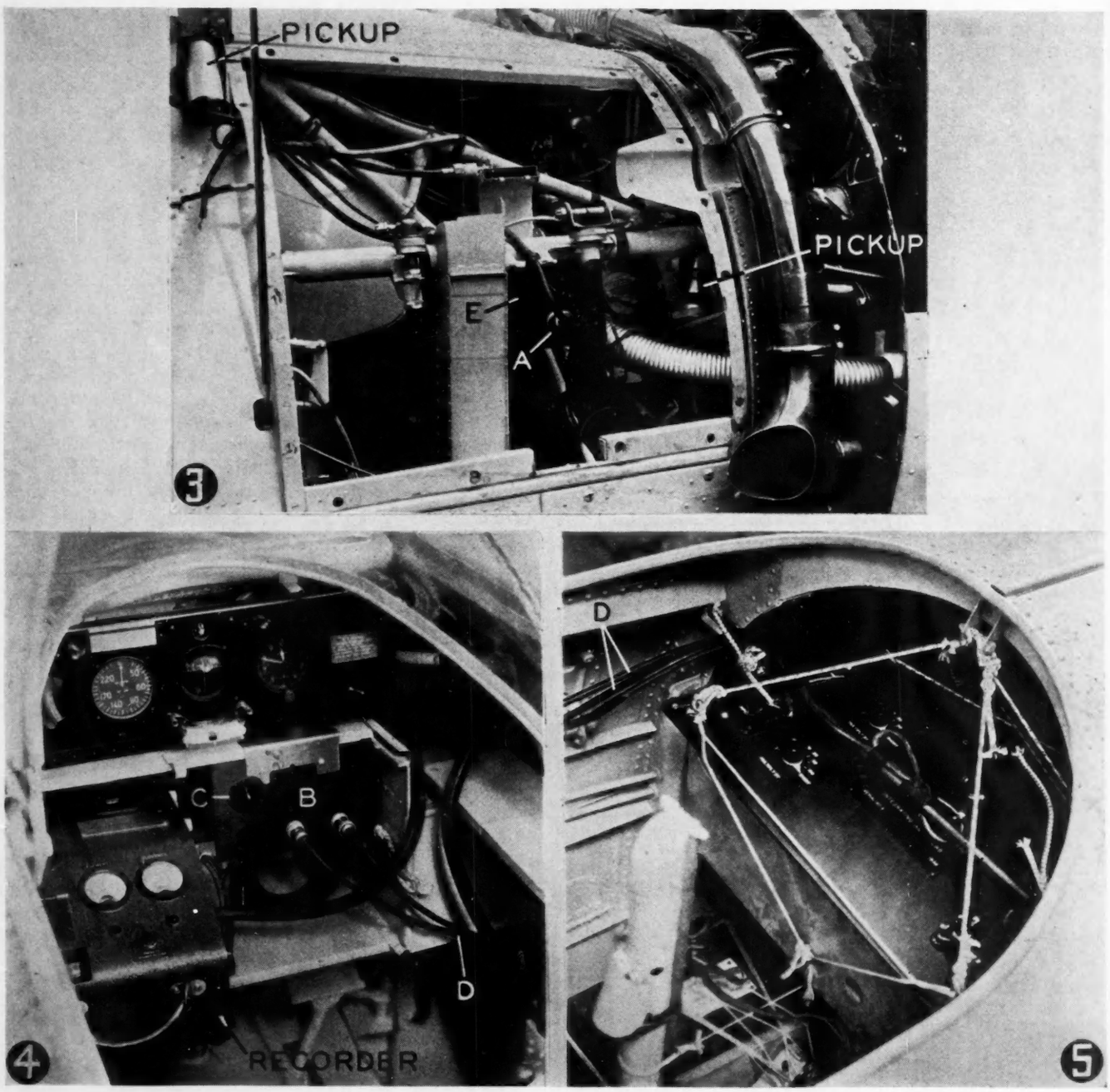


Fig. 3 – Installation of Pickup Units and Distribution Panel in Test Airplane

Fig. 4 – Recording Oscillograph and Control Panel Installed in Airplane Cockpit

Fig. 5 – Amplifiers Installed in Rear of Cockpit

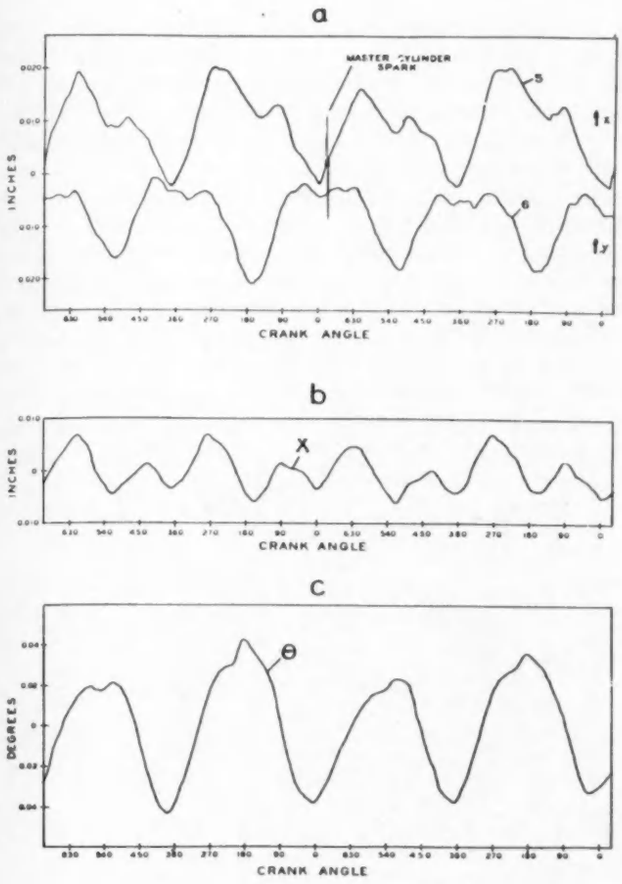


Fig. 6 – Separation of Record Traces into Curves of Torsional and Translational Motion

of the engine. In addition, the fuselage structure has its resonance just above the second-order engine frequency at the upper limit of the operating range. The indicated remedy for these conditions is a somewhat more flexible engine mount and a stiffer fuselage structure. Information of this sort in the hands of the designer will certainly result in airplanes less subject to vibration than present-day types.

Measurement of Vibratory Strains

Vibration failures in structural members are always due to repeated strains which are measured only indirectly by vibration displacements. For this reason electrical pickups for vibratory strains are included in the measuring equipment. The strain gage is actuated by relative motion between two points on the structure under examination.

An interesting application of the strain pickup is the indication of detonation. For this purpose, a gage unit is connected between a cylinder stud and the spark-plug shell of an engine. With this arrangement the high-frequency vibratory strains characteristic of detonation produce an electrical output which can be measured either with an oscillograph or a long-period averaging meter.

Fig. 9 is a plot of the peak voltage output from the strain gage mounted on an engine cylinder vs. the spark-advance angle. A trained observer was unable to detect the "knock" at point *A* and was just able to distinguish the characteristic sound at point *B*.

Pickup Units

Pickups for measuring the amplitude of linear vibration are based on the familiar "seismographic" principle. The seismic element is a mass carried by a supporting element which is rigidly connected to the vibrating member. The seismic element is free to move in one direction with respect to the vibrating member except for a spring coupling and a damping element. The spring is made so weak that, in the operating range of the instrument, the seismic element moves at the average speed of the vibrating member without vibration. With this condition a record of the relative motion between the supporting element and the seismic element will give directly the vibratory displacements to be measured. Fig. 10 shows the application of the seismic principle to the measurement of linear and torsional vibration displacements.

Relative motion between the seismic element and a vibrating member is transformed into electrical voltage impulses by means of small electromagnetic generators incorporated in the pickups. In the linear-motion pickup the seismic element carries a permanent magnet and yoke (Fig. 11, *left*), which maintains flux in a radial air gap. A cylindrical shell passes through the radial gap and carries a winding which receives an induced voltage whenever relative motion occurs between the seismic element and the supporting element. The generator for the torsional-vibration pickup is similar to an ordinary magneto except that the armature oscillates between the

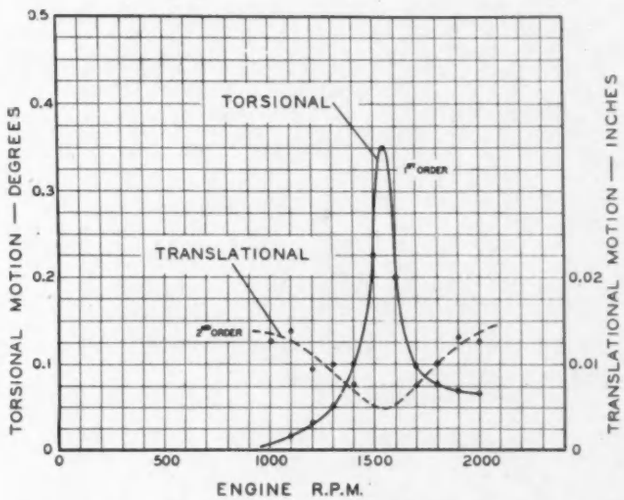


Fig. 7 – Torsional Vibration of Engine Crankcase about Thrust Axis and Translational Span-Wise Vibration of Engine Crankcase in Flight

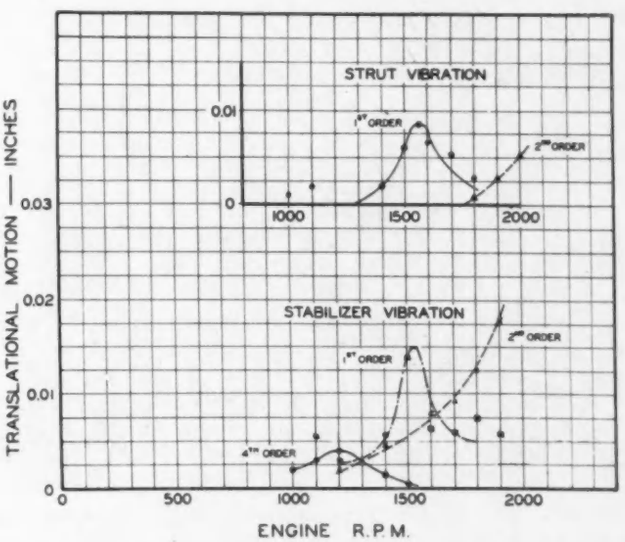


Fig. 8 – Vertical Vibration of Interplane Strut and Stabilizer Root in Flight

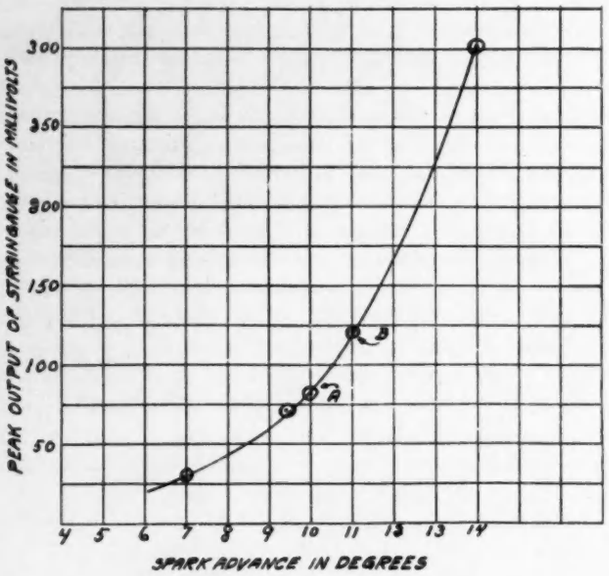
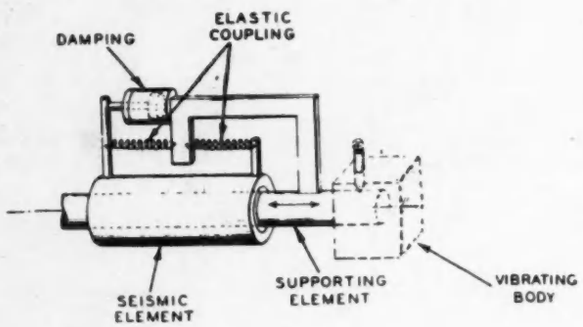


Fig. 9 – Output Voltage of Strain Gage Mounted on Cylinder-Head of Detonating Engine

pole pieces instead of rotating (Fig. 11, *right*). With the pole pieces carried by the seismic element and the armature attached to the shaft, the arrangement produces a voltage at the slip-ring brush terminals whenever the shaft has torsional



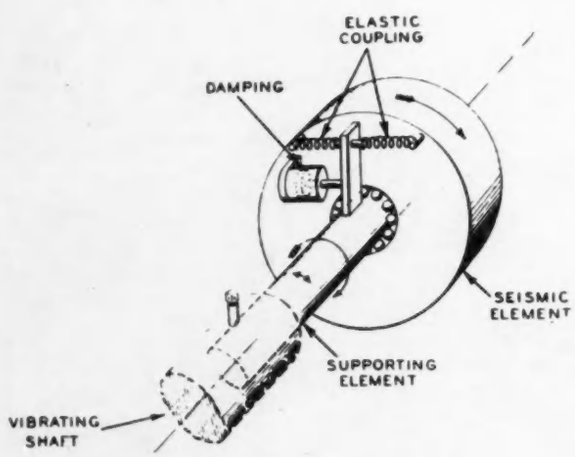


Fig. 10 – Application of the Seismic Principle to Linear and Torsional Vibration Pickup Units

oscillation. The essential features of the electrical generators are shown in Fig. 11.

Damping is required in both types of units to eliminate the effects of natural frequency oscillations in the seismic system. In the linear-vibration pickup this damping is obtained by operating the unit in oil which is pumped in and out of the coil form when relative motion occurs in the system. A bakelite plug carried by the seismic element and forced against the supporting element is used for damping in the torsional-vibration pickup.

The effect of damping on the performance of a linear-vibration unit is shown in Fig. 12. With low damping forces the effect of resonance in the seismic system is apparent as an increased magnification (magnification is defined as amplitude of relative motion/amplitude of vibrating member) just above the natural frequency of the unit. This effect can be eliminated by an increase in damping, but the unit will not operate properly until a frequency equal to two or three times the natural frequency of the seismic system is reached. In any case, rubbing friction between the seismic element and the

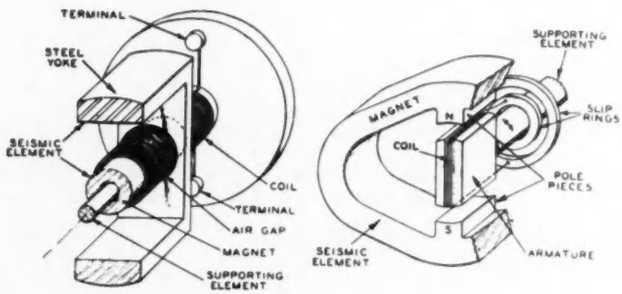


Fig. 11 – Electromagnetic Generators for Linear and Torsional Vibration Pickup Units

supporting element will establish some lower frequency limit where the unit will "freeze up" and cease to operate for a constant amplitude of vibration. In the ordinary airplane-vibration problems of today this rubbing-friction effect does not impose a limitation on the operation of the pickups.

The external appearance of a linear-vibration pickup is shown at *A* in Fig. 13. At *B* a disassembled unit is adjusted for horizontal operation, whereas the springs of the unit *C* are altered for vertical operation. The bakelite cap *S* serves to protect the terminals and to hold the insulated leads in place. In Fig. 14 the unit of Fig. 13 is in the middle with a large pickup for low-frequency work at the left and a small unit for light airplane structures at the right. The weights of the units

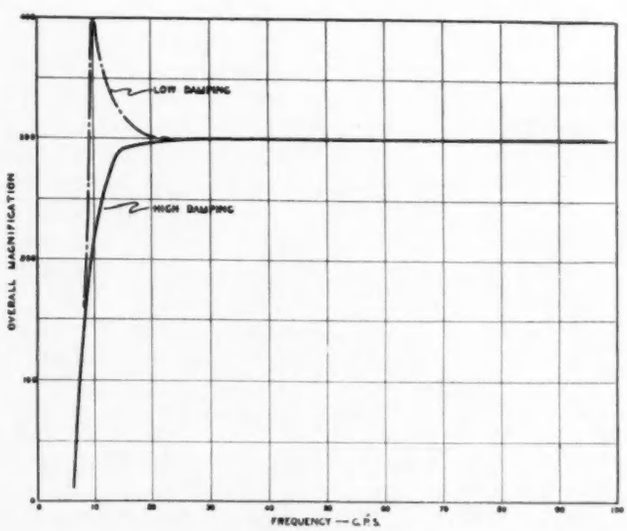


Fig. 12 – Effect of Damping in Pickup Unit on Overall Response of Measuring Equipment

are 3 lb., 1 lb., and 3 oz. in the order of decreasing size (left to right).

A torsional-vibration pickup is shown at the left of Fig. 15 with the various working parts displayed at the right. The seismic element is a brass cylinder, *A*, which carries the magnets, *B*, and the pole pieces, *C*. The steel flange, *D*, is designed to be rigidly connected to the shaft whose vibration is being recorded. The supporting element, *E*, is clamped to the flange by the nut, *F*, and carries the armature, *N*, which fits between the pole pieces, *C*, when the unit is assembled. The coupling springs are connected between pins on the rear faces of the seismic element and the mounting flange. Slip-rings are set into the bakelite discs on the supporting element and make contact with the terminals, *R*, through small carbon brushes. An aluminum cylinder, *Q*, carries the terminals which are connected to a flexible double-conductor cable clamped under the bakelite cap, *U*. The unit has a weight of about 3¼ lb.

Strain Gage

Vibratory strains are measured with the aid of a pickup for transforming small motions into voltage changes proportional to the velocity of motion. Fig. 16 shows the essential parts of the strain pickup, which incorporates suggestions due to Dr. W. M. Bleakney of the Bureau of Standards. Magnet *A* and yoke *B* maintain flux in cylindrical iron armatures, *C*, which carry coils of fine wire, *D*. These coils are connected in series to the terminals, *E*. The armatures are freely suspended in

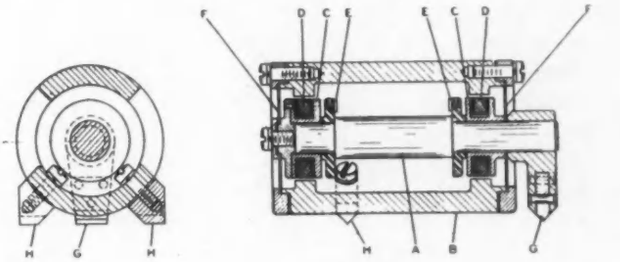
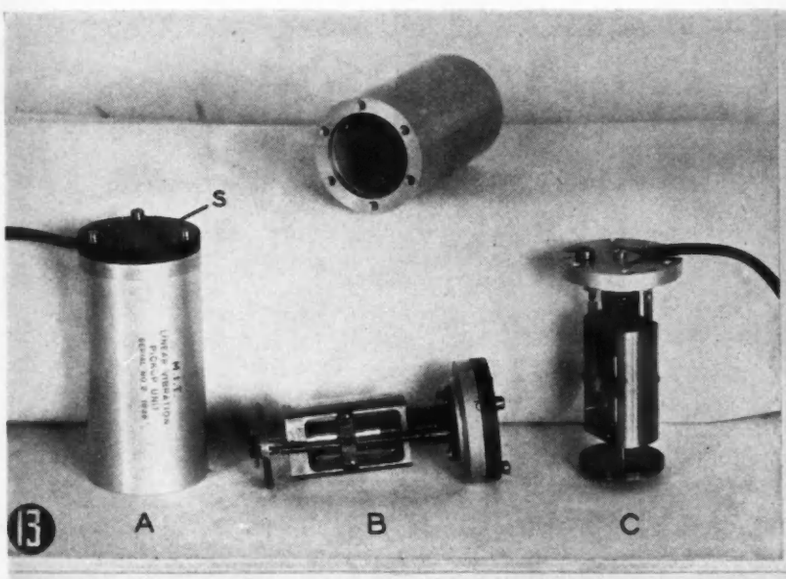
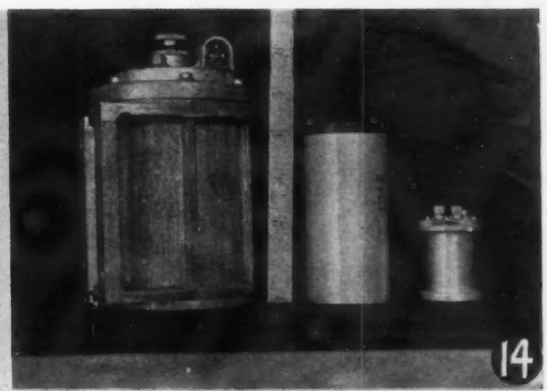
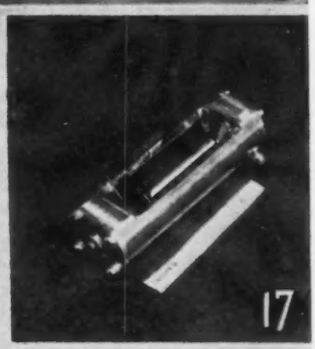
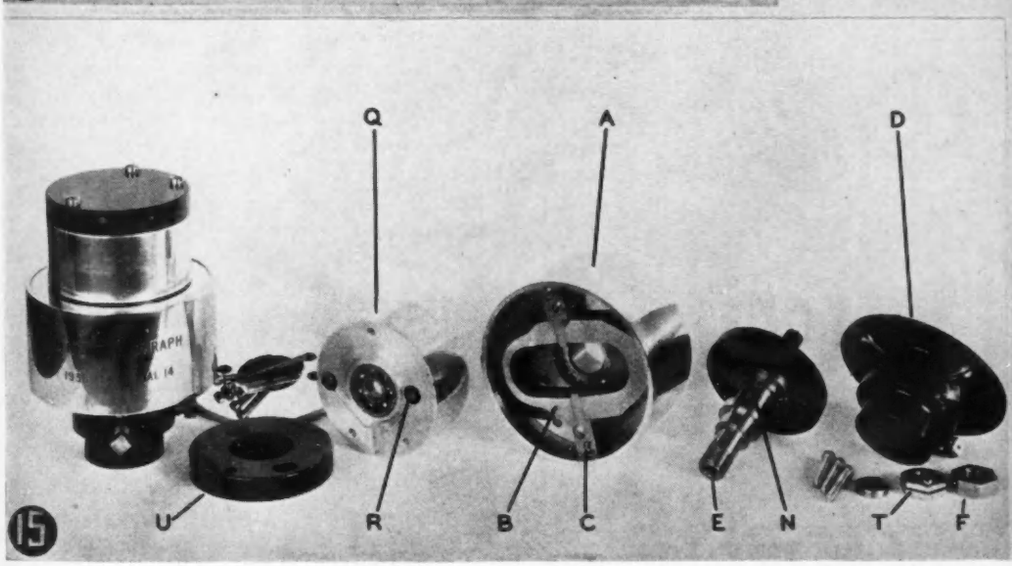


Fig. 16 – Pickup for Measuring Vibratory Strains









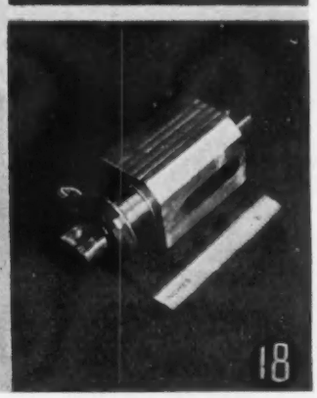


Fig. 13 – Linear-Vibration Pickup Unit

Fig. 14 – Three Types of Linear-Vibration Pickup Units

Fig. 15 – Torsional-Vibration Pickup Unit Assembled and Disassembled

Fig. 17 – Pickup Unit for Structural Vibratory Strain

Fig. 18 – Pickup for Detonation Strain

the gaps by means of thin diaphragms, *F*. The armature unit is clamped to the member under test through knife edge, *G*, at one end of the gage length, while a point on the yoke is clamped at the other end of the gage length by the knife edge, *H*. Any strain in the member which is accompanied by a displacement between the knife edges will cause a shift in flux from one side to the other of each armature and produces a voltage in the coils proportional to the rate of displacement. The strain pickup has a weight between 2 and 3 oz. depending on the gage length chosen. When used with the standard amplifier and oscillograph, displacements on a specimen are magnified about 1000 times.

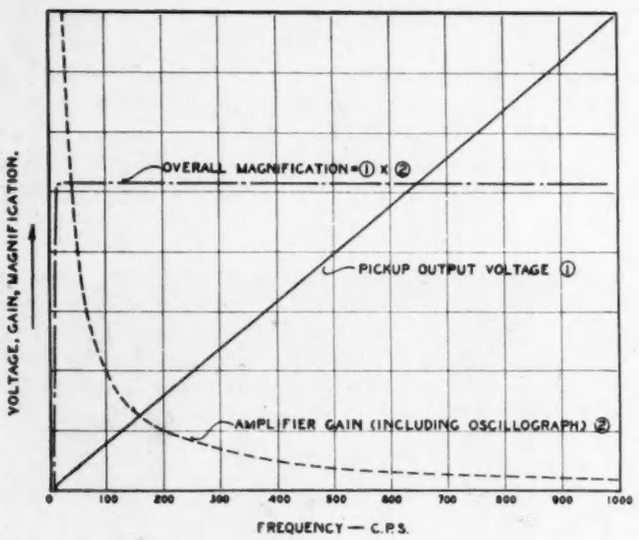


Fig. 19 – Combination of Pickup and Amplifier Characteristics Yielding Overall Output Independent of Frequency

Fig. 17 shows a strain gage having knife edges for a base line of 2 in. Fig. 18 is a photograph of a strain gage adapted to measure detonation strains in an engine cylinder-block.

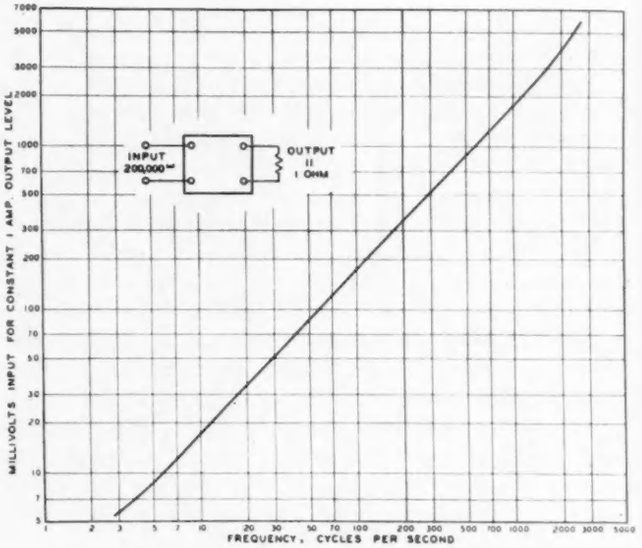


Fig. 21 – Sensitivity Curve for Integrating Amplifier

Amplifier

The pickups described previously have the common characteristic that the voltage output is proportional to the velocity of motion. For a constant amplitude of sinusoidal vibration this relation means that the voltage amplitude will increase directly with frequency as shown in the straight line (1) of Fig. 19. In order to produce the desired result of an output which is independent of frequency, the coupling amplifier must have a sensitivity which decreases with increasing frequency. The necessary amplifier characteristic is indicated in curve (2) of Fig. 19, which also shows the product of ordinates of (1) and (2) as a straight line of constant ordinate which is the desired overall frequency response. An amplifier which gives an output proportional to the amplitude of vibration when the input is proportional to velocity, actually performs the function of integration and is therefore called an "integrating amplifier."

The wiring diagram of an "integrating amplifier" designed to operate a conventional electromagnetic oscillograph is

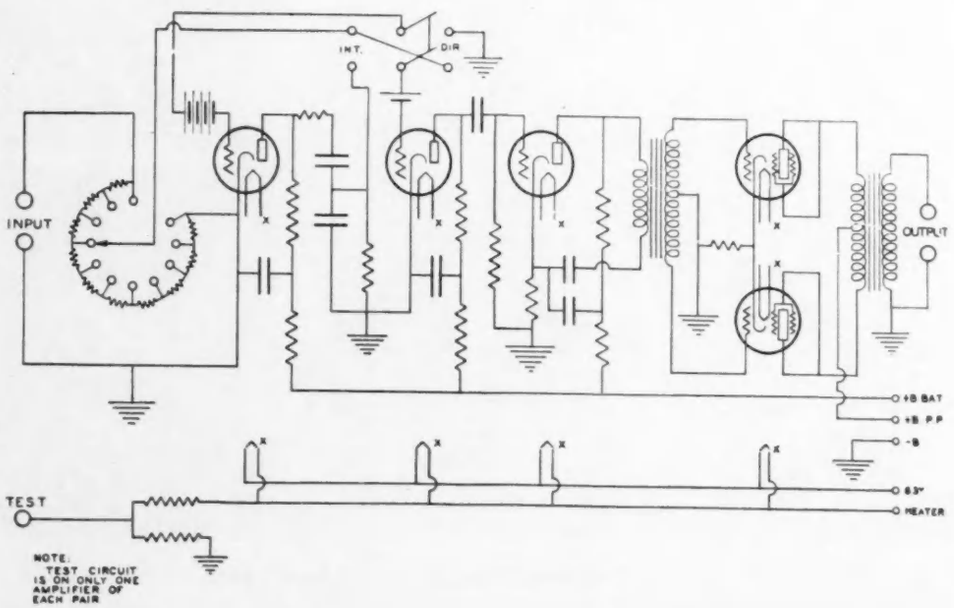


Fig. 20 – Schematic Circuit Diagram of Integrating Amplifier

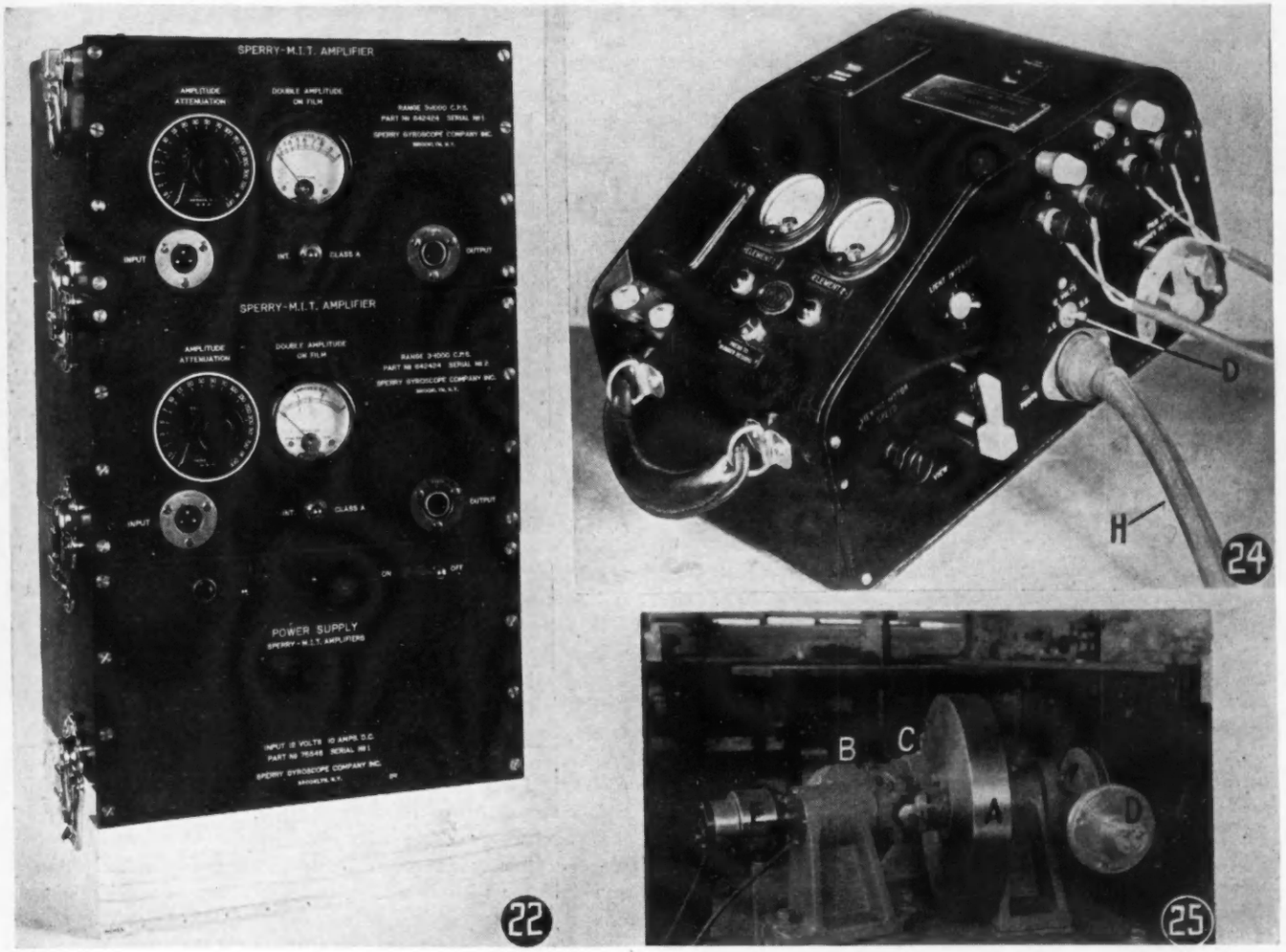


Fig. 22 – Two-Channel Amplifier for Airplane Flight-Test Work
Fig. 24 – Recording Oscillograph Showing Controls
Fig. 25 – Calibrator for Vibration-Pickup Units

shown in Fig. 20. The integrating circuit consists of a resistance in series with two condensers connected across the first-stage load resistor. The elements are chosen so that the current in this circuit is proportional to the applied a.c. voltage over the operating range of frequencies. Under this condition the voltage across each condenser will be proportional to the time integral of the a. c. voltage across the load resistor. The integrated voltage is applied to the grid of the second tube by a connection between the two condensers. This arrangement serves to isolate the grid of the second tube from the plate voltage of the first tube. A switch removes the integrating circuit when it is desired to use the unit as a straight amplifier.

Fig. 21 shows the sensitivity curve for the integrating amplifier. The input voltage required to give a fixed output increases directly with frequency from about 5 cycles per sec. to 1500 cycles per sec.

Fig. 22 is a photograph of a two-channel amplifier with power supply adapted for use in small airplanes. Its design is such that any number of amplifier channels and power supplies can be racked together to meet the requirements of a particular problem. The weight of the two-channel amplifier and power supply is 62 lb.

Oscillograph

Fig. 23 is a schematic diagram of the recording oscillograph. The instrument is designed to be operated by a single

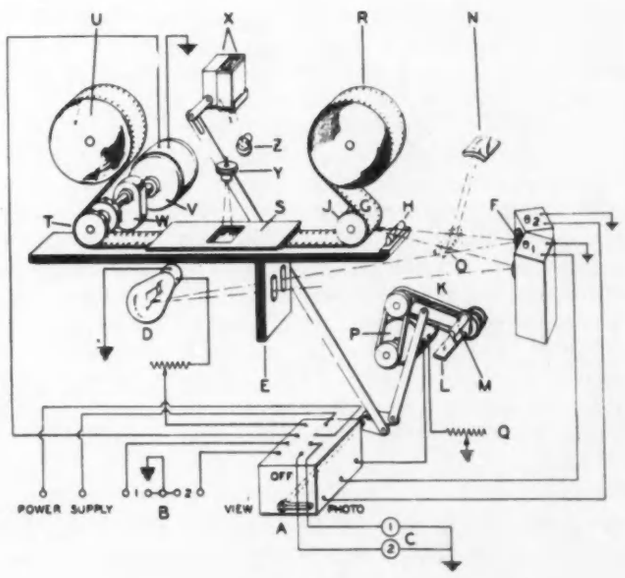


Fig. 23 – Schematic Diagram of Recording Oscillograph

control lever, *A*, which connects the two independent sets of input terminals, *B*, to the meters, *C*, when in the "off" position. When the control lever is turned to the "photo" position, the ribbon filament lamp, *D*, is lighted and the oscillograph elements, 1 and 2, are connected to the terminals, *B*. Light from *D* passes through openings in the baffle, *E*, and is reflected from mirrors, *F*, in the elements 1 and 2. The light beams are focused to a spot on the film, *G*, over the roller, *J*, by means of the cylindrical lens, *H*. When it is desired to examine the vibration visually, the lever, *A*, is moved to the "view" position which raises the unit, *K*, until the mirror, *L*, intercepts the light beams as shown in the dotted position at *O*. The motor, *P*, whose speed can be adjusted by means of the rheostat, *Q*, rotates mirror, *L*, about its long axis. At the same time the cylindrical lens, *M*, mounted parallel to the mirror is revolved so that it focuses the beams as they are swept across the screen, *N*. Film is supplied from a standard roll of motion-picture stock shown at *R*. From the roller, *J*, the film is carried under a masking gage, *S*, over a sprocket, *T*, and on to the take-up reel, *U*. The sprocket is driven by a motor, *V*, through a change-gear box, *W*. This motor is started when the control lever is moved to the "photo" position. The movement of the control lever to make a record automatically advances the two counters, *X*, by one number. After a record is completed, the number is photographed on the film by means of the small lamp, *Z*, and the lens, *Y*.

The recording oscillograph can be operated on either alternating or direct current. The only adjustment necessary is to shift a switch to the properly marked side. No auxiliary means for timing the records is required since the film-drive motor is governor-controlled to have a constant speed. The instrument can be loaded in daylight and the exposed film removed in daylight.

Fig. 24 is a photograph of the recording oscillograph with connections made for use. Either a.c. or d.c. power at 12 volts is supplied through the cable, *H*. The a.c.-d.c. selector switch is shown at *D*. The other controls can be identified by labels on the instrument itself. The recording oscillograph has a weight of 38 lb., is 14 in. long, 10 in. wide and the greatest height is 8 in.

Calibrator

Overall calibrations of the equipment are carried out by means of the unit shown in Fig. 25. A heavy flywheel, *A*, is driven at constant speed from a synchronous motor, *B*. The speed can be adjusted by means of a variable-speed transmission, *C*. To calibrate a linear-vibration pickup, the unit is mounted on an eccentric-driven follower, as at *D*. The frequency of vibration is equal to the speed of the flywheel. A double-eccentric arrangement makes it possible to adjust the amplitude of motion from 0.00 to 0.050 in. Torsional vibration of known amplitude is obtained by means of a universal joint with the driven shaft set at a small angle to the driving shaft. The vibration will be a pure sine wave for practical purposes if the shaft angle is not greater than 10 deg. A torsional-vibration pickup unit mounted for calibration is shown at *E*. The calibrator has a range of 0 to 100 cycles per sec. in linear vibration and twice this range in torsional vibration since two cycles are completed for each revolution of the shaft.

Conclusions

Several months of use in the hands of manufacturing and test organizations have shown that the equipment described previously will give reliable results both in the test house and in actual flight. The data already obtained supply a basis for determining the steps necessary to solve a variety of vibration problems in aircraft.

It is to be expected that a systematic application of the apparatus will yield an increasing amount of information which will allow worthwhile advances in the art of aeronautics in the direction of more comfortable and safer aircraft.

Principle of the Sunbury Knock Indicator

RESEARCH on the effects of detonation on the C.F.R. engine has shown that the cylinder strains, measured between the top and a point near the lower end of the cylinder casting, increase as the intensity of detonation increases. Examination of cathode-ray oscillograms of the strains, and also of diagrams showing the velocities at which the various vibrations occur, has indicated that the mean value of the *velocity* of vibration is a very good measure of the effect on the ear of the knock noise.

It is unnecessary here to enter into a detailed account of the causes of this behavior, although it may be mentioned that a reasonable explanation can be given of the agreement which is found to exist.

A very convenient method for measuring the velocity at which the cylinder is strained depends on fixing an iron rod to the upper part of the cylinder and mounting an electromagnetic-type pickup unit on the water connection at the lower end of the casting, the lower end of the rod and the pole at the pickup being adjusted to give an air gap of a few thousandths of an inch. The pickup unit consists of a permanent magnet around the pole of which is wound a coil of fine wire connected to terminals from which the electrical output may be taken, this output being proportional to the velocity at which the air gap alters. Leads are carried from the unit to the grid of a rectifying and amplifying vacuum tube and thence to a knockmeter. The reading on the meter can be arranged to vary with knock intensity.

The mechanical construction necessary for vibration measurements of this type may be very complicated on certain engines, especially when they are more or less flexibly mounted in an automobile chassis or when the construction is light and highly stressed, but it has been proved by tests both with more elaborate apparatus and with the simple system just mentioned that there is no advantage to be gained by any special elaboration in the case under consideration. It is desirable, however, that the pickup unit itself should be so made that it does not suffer from effects due to internal vibrations caused by engine detonation, and it should, of course, be suitable for continuous use under ordinary routine testing conditions.

Among the advantages of a knock indicator built on these lines are:

(1) It has no moving parts other than the pointer of a knockmeter.

(2) The components mounted on the engine are not subject to gas pressure or temperature, but are rigidly fixed to the cylinder and are external to the combustion space.

(3) No major alteration is necessary to equip existing C.F.R. engines with the apparatus, which can be fitted in a few minutes.

(4) Adjustments are not critical and alteration of the air gap only affects sensitivity without altering ratings.

(5) Readings alter very little throughout the day on a given fuel at a given compression ratio, provided no important alteration in humidity or barometric pressure occurs.

Excerpt from the paper: "The Sunbury Knock Indicator," by E. S. L. Beale and R. Stansfield, presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 7, 1937.